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USAAVLABS TECHNICAL REPORT 68-83

HELICOPTER TRANSMISSION OIL HEAT REJECTION INVESTIGATION

By

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John D. Buteau

May 1969

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

CONTRACT DAAJ02-67-C-0077

SIKORSKY AIRCRAFT
DIVISION OF UNITED AIRCRAFT CORPORATION
STRATFORD, CONNECTICUT

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172

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This Command concurs in the conclusions stated in the report.

In view of the significant reduction of vulnerability that can be obtained, an integral type of cooling system for the main transmissions should be considered in the design of future Army helicopters.

Task 1G162203D14415
Contract DAAJ02-67-C-0077
USAAVLABS Technical Report 68-83
May 1969

HELICOPTER TRANSMISSION OIL HEAT REJECTION INVESTIGATION

Sikorsky Engineering Report 50558

By

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for

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SUMMARY

This report presents the results of a feasibility and preliminary design study of integral cooling systems for helicopter main transmissions. The purpose of the study is to provide a method or methods whereby the requirements for remote external cooling systems no longer exist. The goal of the program is to define long-range solutions and/or more expedient methods permitting simple retrofit to existing aircraft.

Emergency methods of gearbox lubrication and cooling, designed specifically to increase the survivability time of transmissions in the event of small-arms projectile damage, are being investigated under other programs and are not included in this effort. The systems evaluated herein are for full-time gearbox lubrication and cooling and as such will function under all operating conditions.

While the primary effort of this study centers around the heat rejection requirements of the main gearbox of the U. S. Army CH-54A crane helicopter, the adaptability of the most promising systems to other U. S. Army helicopters is also investigated.

During this study, several extremely advanced methods of gearbox cooling were investigated. Although the thermodynamic principles of those concepts are sound, current state-of-the-art hardware makes the systems impractical because of weight and effective power requirements, which result in significant degradations in aircraft performance. Two of the systems investigated, the rotor shaft and oil sump coolers, present relatively efficient and invulnerable solutions to the problem. These systems are substantiated by detailed analysis, empirical data, and consultation with many heat exchanger manufacturers.

While the rotor shaft and oil sump cooling systems are found to be more efficient, lighter, and less vulnerable than systems employing an intermediate heat transfer fluid, a vapor cycle refrigeration cooling system is also evaluated. While the vulnerability of the oil circuit of this design can be reduced to that of the integral oil-to-air systems, the vulnerability of the refrigerant-to-air circuit and the lower efficiency of heat rejection make this system less desirable than the oil-to-air systems presented.

Detailed reliability, maintainability, and vulnerability analyses of the three systems examined in depth are presented in comparison to the present CH-54A external cooling system. In addition to a failure mode and effect analysis, estimated maintenance man-hours and MTBFs are presented for the integral cooling and current external cooling systems.

Both the rotor shaft cooling system and the oil sump cooling system are within current state-of-the-art hardware development and are adaptable to several helicopters currently in the U. S. Army inventory. On the basis of this preliminary investigation, this adaptation can be accomplished on the CH-54A, CH-47A, and UH-1D with a modest amount of design change and development effort.

FOREWORD

This report covers a feasibility and preliminary design study of integral cooling systems for helicopter main transmissions to reduce the vulnerability of these systems to small-arms fire. The study was conducted during the period from July 1, 1967 through July 31, 1968, for the U. S. Army Aviation Materiel Laboratories (USAAVLABS) under Contract DAAJ02-67-C-0077 (Task 1G162203D14415). Pertinent empirical data and analysis used to substantiate some portions of this report were provided by the following: United Aircraft Products, Incorporated, Dayton, Ohio; Lytron Incorporated, Woburn, Mass.; Janitrol Aero Division, Midland-Ross Corporation, Columbus, Ohio; Joy Manufacturing Co., Pittsburgh, Penn.; and Benson Manufacturing Division, Electronic Communications, Incorporated, Kansas City, Mo.

USAAVLABS technical direction was provided by W. A. Hudgins of the Mechanical Systems Branch, Aircraft Systems and Equipment Division.

The principal investigators for Sikorsky Aircraft were L. R. Burroughs, Supervisor, Mechanical Systems Section; D. A. Storm, Sr. Design Analytical Engineer; and J. D. Buteau, Design Engineer of the Transmission Group. Also making significant contributions to this effort were J. Ellingsen of the Transmission Design Group and P. Beaver of the Maintainability and Reliability Section.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

Figure		Page

LIST OF TABLES

LIST OF SYMBOLS

a_o	flow factor, oil-side pressure head
acc	acceleration
act	actual
A	heat transfer area, ft^2
A_a	unit air-side heat transfer area, ft^2/ft
A_c	convection surface area, ft^2
A_{cs}	air-side cross-sectional area, $in.^2$
A_o	unit oil-side heat transfer area, ft^2/ft
A_{os}	minimum oil-side free-flow area, $in.^2$
A_r	radiative surface area, ft^2
A_w	mean heat transfer area, ft^2
AHP	air horsepower, HP
AN	Army - Navy
APU	auxiliary power unit
b_t	baffle spacing, in.
B_o	empirically determined flow reversal factor
B-10	bearing life that 90 percent of the population will exceed
c_p	specific heat of fluid at constant pressure, Btu/lb^{o}F
comp	compressor
cond	condenser
cos	cosine
C	flow-stream capacity rate, Btu/min^{o}F

C_{max}	oil flow-stream capacity rate, Btu/min°F
C_{min}	airflow stream capacity rate, Btu/min°F
D_h	(equivalent) hydraulic diameter, in.
D_i	air tube inside diameter, in.
D_{i_m}	transition duct mean inside diameter, in.
D_o	air tube outside diameter, in.
DN	product of bearing bore times shaft speed, in./rpm
e	heat exchanger core effectiveness, percent
evap	evaporator
E	Young's modulus of elasticity, psi
EHD	elastohydrodynamic
f	coefficient of friction for airflow
f_d	drag coefficient of friction for air flowing in smooth tubes (not Darcy-Weisbach friction)
f_o	coefficient of friction for oil flow
foul	fouling
F	flow-stream capacity rate ratio
FHP	friction horsepower, HP
g	Newtonian gravitational constant, ft/sec^2, ft/hr^2
gpm	gallons per minute
G	flow-stream mass velocity, lb/hr ft^2
G_a	air-stream mass velocity, lb/hr^2
G_o	oil-stream mass velocity, lb/hr ft^2
h	enthalpy, Btu/lb

$\bar{h}$	film heat transfer coefficient, Btu/hr ft^2 °F
$\bar{h}_a$	air-side heat transfer coefficient, Btu/hr ft^2 °F
$\bar{h}_{a_{foul}}$	air-side heat transfer coefficient for fouled walls, Btu/hr ft^2 °F
$\bar{h}_o$	oil-side heat transfer coefficient, Btu/hr ft^2 °F
$\bar{h}_{o_{foul}}$	oil-side heat transfer coefficient for fouled walls, Btu/hr ft^2 °F
HP_{comp}	compressor theoretical horsepower
int	internal
j_a	air-side generalized heat transfer grouping, dimensionless
k	thermal conductivity, Btu/hr ft °F
kw	kilowatt
K	linear proportionality constant
K'	linear proportionality constant
K_c	sudden contraction loss coefficient
K_e	sudden expansion loss coefficient
K_{el}	elbow loss coefficient
L	heat exchanger core length, ft
L_e	equivalent straight line elbow length, in.
LMTD	log mean temperature difference
m	gear ratio
$\dot{m}$	mass flow rate, lb/min
$\dot{m}_a$	mass flow rate of air, lb/min

$\dot{m}_o$	mass flow rate of oil, lb/min
$\dot{m}_r$	mass flow rate of refrigerant, lb/min
max	maximum
misc	miscellaneous
MH	man-hours
MR	main rotor
MS	military standard
MTBF	mean time between failures
N	total number of air tubes
N_b	number of air tubes at baffle leading edge
NTU	number of (heat) transfer units
ppm	parts per million
P	pressure head, suction head, lb/ft^2
P_{cond}	head through condenser, lb/ft^2
P_{core}	head through heat exchanger core, lb/ft^2
$P_{flow\ acc}$	head due to flow-stream acceleration, lb/ft^2
P_{total}	total head, lb/ft^2
P_r	Prandtl number, a fluid properties modulus, dimensionless
Press.	pressure
$\dot{Q}$	oil heat rejection rate, Btu/min
$\dot{Q}_c$	convection heat rejection rate, Btu/min
$\dot{Q}_r$	radiative heat rejection rate, Btu/min
$\dot{Q}_t$	total heat rejection rate, Btu/min

$\dot{Q}_{max}$	maximum possible oil heat rejection rate through a perfect heat exchanger, Btu/min
R	universal gas constant, ft/°R
R_s	shaft radius, in.
R_e	Reynolds number, a flow modulus, dimensionless
R_{e_a}	air-side Reynolds number, dimensionless
R_{e_o}	oil-side Reynolds number, dimensionless
s	entropy, Btu/lb °R
s_t	oil-side free-flow spacing between air tubes, in.
St	Stanton number, a heat transfer modulus, dimensionless
SAE	Society of Automotive Engineers
SHP	shaft horsepower, HP
SHP_{comp}	compressor shaft horsepower, HP
$SHP_{cond\ blower}$	condenser blower shaft horsepower, HP
SHP_{total}	total shaft horsepower, HP
T	temperature, °F, °R
T_a	ambient air temperature, °F
T_{act}	actual temperature, °F
T_{a_1}	air inlet temperature, °F
T_{a_2}	air outlet temperature, °F
$T_{int_{cw}}$	internal gearbox temperature at current production operating temperature, °F
$T_{int_{high}}$	internal gearbox temperature at proposed raised operating temperature, °F

T_{max}	maximum temperature, °F
T_{o_1}	oil inlet temperature, °F
T_{o_2}	oil outlet temperature, °F
T_s	gearbox skin temperature, °F, °R
$T_{skin_{low}}$	external gearbox skin temperature at current production operating temperature, °F
$T_{skin_{high}}$	external gearbox skin temperature at proposed raised operating temperature, °F
TR	tail rotor
Trans	transmission
U	overall heat transfer coefficient, Btu/hr ft^2 °F
UA_{reqd}	product - overall heat transfer coefficient times heat transfer area required, Btu/hr °F
UA/ft	unit product - overall heat transfer coefficient times heat transfer area, Btu/hr ft °F
V_1	inlet velocity, ft/sec
V_2	outlet velocity, ft/sec
V_m	mean velocity, ft/sec
W	total vector load, lb
x_t	ratio of transverse tube pitch to tube outside diameter
x_w	air tube wall thickness, in.
°F	degrees Fahrenheit
°R	degrees Rankine
β_a	arc of approach, involute gearing, radians
β_r	arc of recess, involute gearing, radians

Δ	differential, difference
ϵ	emissivity, dimensionless
η	efficiency, percent
μ	viscosity, lb sec/in.^{2}, lb/hr ft
ν	Poisson's ratio
ρ	density, lb/ft^3
ρ_1	density at inlet, lb/ft^3
ρ_2	density at outlet, lb/ft^3
ρ_m	mean density, lb/ft^3
σ	Stefan-Boltzmann constant, Btu/hr ft^2 ^oR
Σ	summation
ϕ	pressure angle, involute gearing, deg
ϕ_n	normal pressure angle, involute gearing, deg
ψ	spiral angle, involute gearing, deg
ω	ratio of free-flow area to frontal area
γ	specific-heat ratio

INTRODUCTION

The lubrication systems of current production helicopter transmissions rely primarily on external sources for cooling, making the transmissions extremely vulnerable to damage from .30 and .50 caliber small-arms projectiles. The elimination of the external lubricant/cooling system will greatly improve helicopter survivability in combat zones.

The current design practice for large helicopter power transmission systems is to dissipate approximately 80 percent of the main gearbox power losses by means of an externally mounted oil-to-air heat exchanger. While other aircraft might take advantage of ram air for cooling, the helicopter power train cooling requirements are critical in hover, where near-maximum gearbox power losses must be dissipated at zero air velocity.

To evaluate the feasibility of eliminating remote external cooling systems for helicopter main transmissions, Sikorsky Aircraft has selected the CH-54A crane helicopter as the aircraft for a conceptual design study. The CH-54A was selected for this study for the following reasons:

- The CH-54A has a transmission rating of 6600 HP, requiring under current design practices an external cooling system heat rejection capability of 5109 Btu/min, per Table I. With its current installed power capability of 8100 HP (4050 HP per engine), this aircraft has excellent single-engine survival capability, which could be further enhanced by reducing the main transmission external cooling system vulnerability.

- Empirical cooling data are readily available on CH-54A transmissions under actual hot-day hovering conditions.

Included in the report are detailed evaluations of several integral oil-to-air systems and a vapor cycle refrigeration system, all of which evolved from a trade-off study of many cooling concepts. Also included is a discussion and an analysis of an oil-fog lubrication system.

BASIC DATA

DESIGN CHARACTERISTICS

Aircraft Configuration

The CH-54A is a single-main-rotor helicopter with an antitorque tail rotor. Two JFTD-12A-4A free turbine engines power the 72-foot-diameter main rotor, giving the aircraft a maximum gross weight of 42,000 pounds and a maximum payload of approximately 18,800 pounds. The primary function of the CH-54A is the transportation of equipment, cargo, and pods suspended below the aircraft. The general arrangement of the CH-54A is shown in Figure 1.

Drive Train Configuration

The production CH-54A power transmission system consists of two engines driving the dual inputs of the main gearbox at 9000 rpm. The main gearbox in turn drives the main rotor shaft at 185 rpm and the tail rotor drive shaft at 3016 rpm. The tail rotor drive system consists of six shaft sections transmitting power to the intermediate gearbox. The intermediate gearbox then reduces the drive speed, changes the drive direction, and powers the pylon shaft and, in turn, the tail gearbox. Another speed reduction and change of drive direction in the tail gearbox brings power to the tail rotor. The overall power transmission system is shown in Figure 2.

The main gearbox, Figure 3, has an input section driven by each of the engines. High-speed spiral bevel sets receive power from the engines, reduce the speed, and change the drive direction by 90 degrees. A second pair of bevel pinions, driven by the output of the first gear set, drives the main bevel gear, which then transmits power to the two planetary stages and the main rotor shaft. The main bevel gear also drives the tail takeoff bevel gear, providing power to the tail rotor drive system. The accessories are driven by a pair of spur gears on the tail takeoff bevel gear shaft.

The current production main gearbox cooling system, Figure 4, consists of a high-speed blower and radiator assembly mounted on the rear cover of the main gearbox. The blower, driven at 4500 rpm by V-belts from a pulley on the tail rotor takeoff flange, delivers 7300 cfm through the transition duct and radiator. As shown in Figure 5, gearbox lubricating oil is taken from the sump through a low-pressure screen by a 36-gpm main shaft driven gear pump, delivering 17.5 gpm to the lubrication system. From the pump, heated oil flows through a 40-micron filter and a 1.25-inch-diameter external flexible line to the radiator. A 1.00-inch-

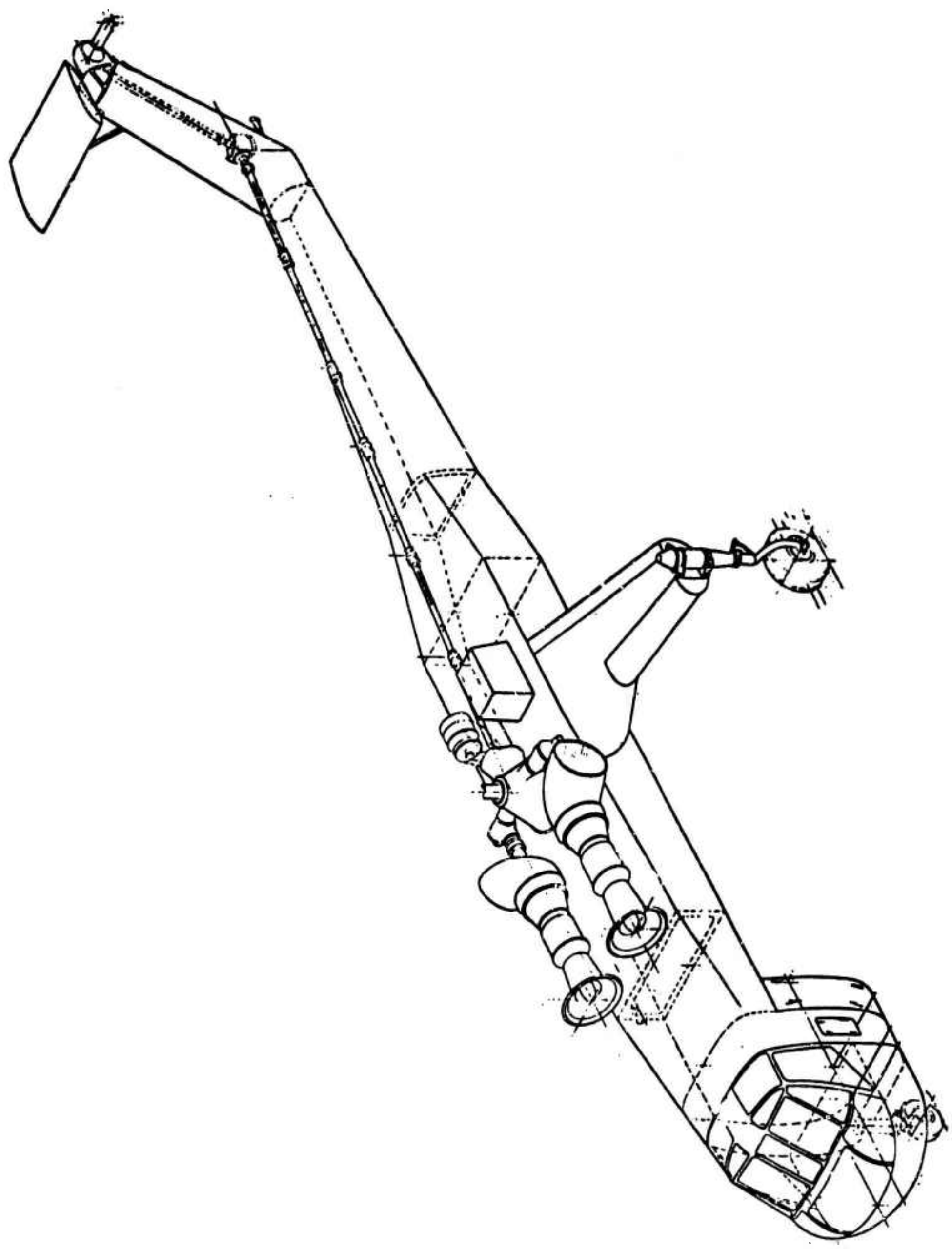


Figure 1. CH-54A Aircraft General Arrangement.

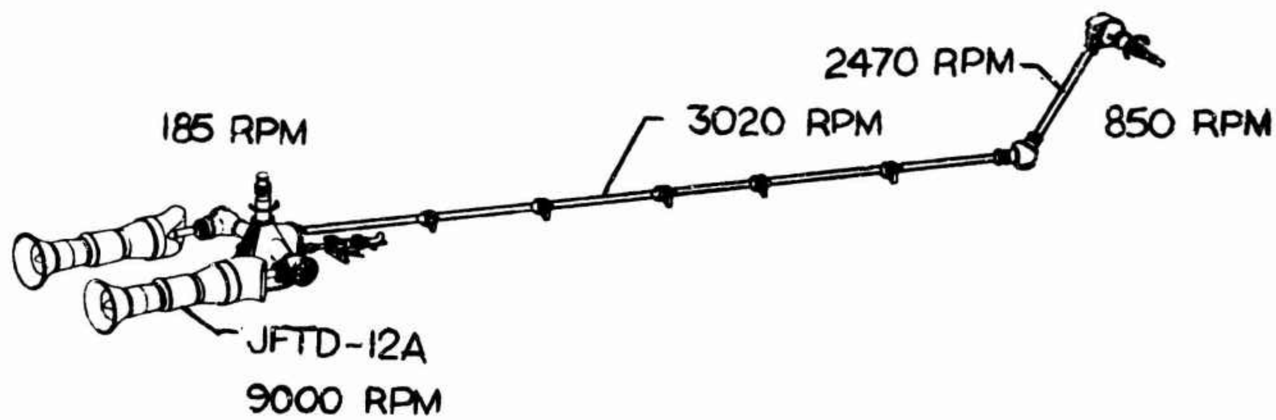


Figure 2. CH-54A Power Transmission System.

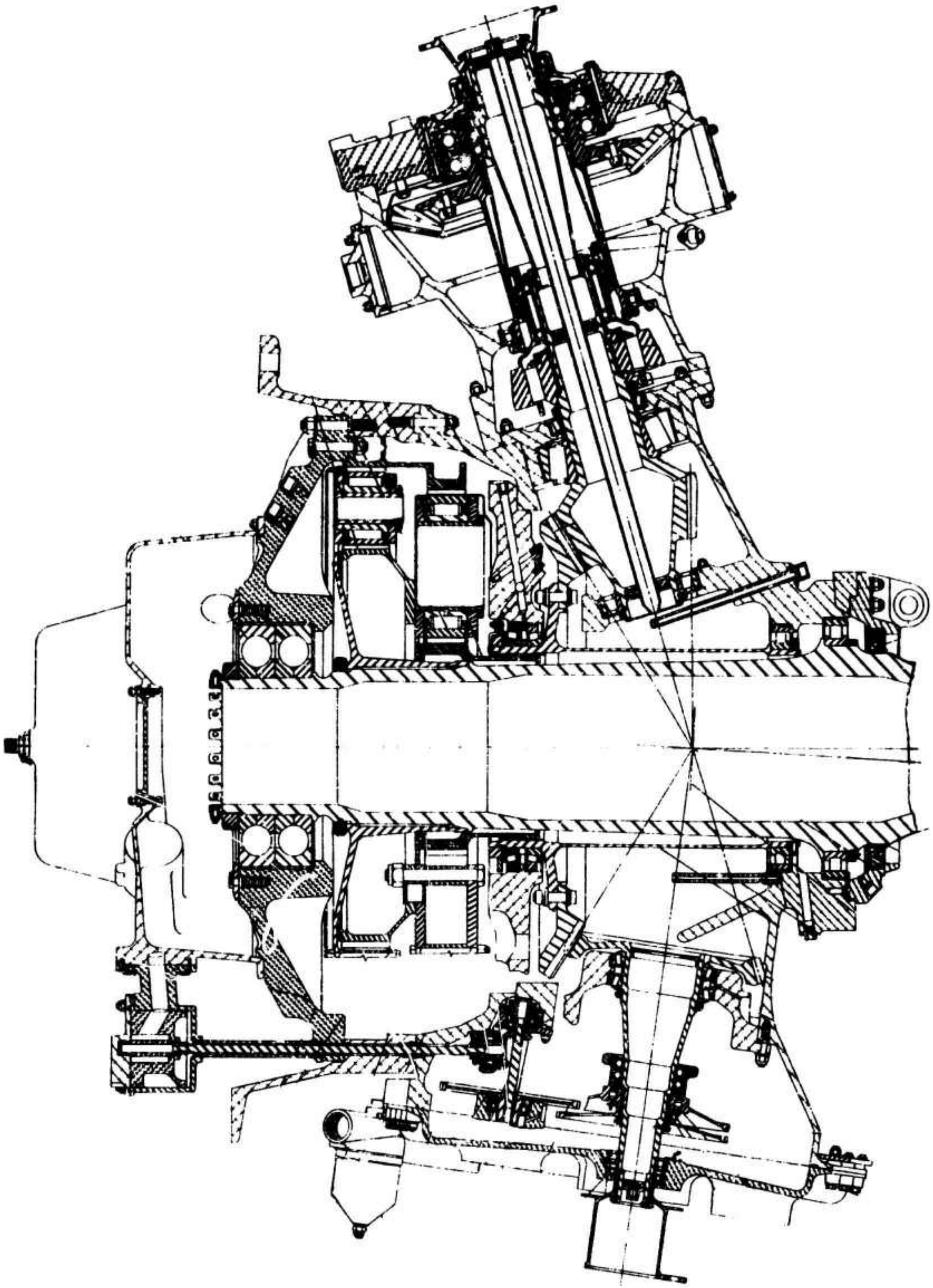


Figure 3. CH-54A Main Gearbox.

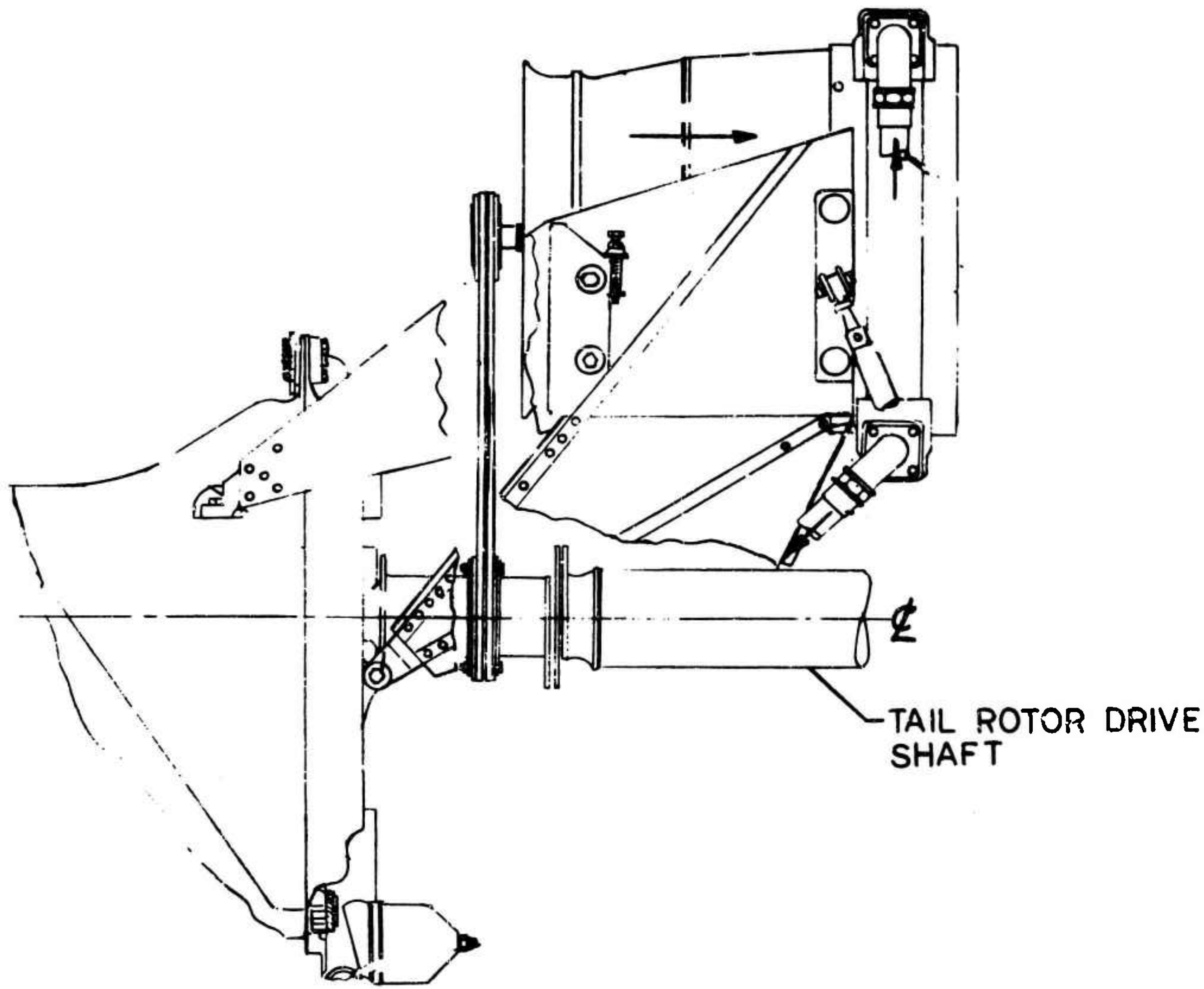


Figure 4. CH-54A Oil Cooler and Blower Installation.

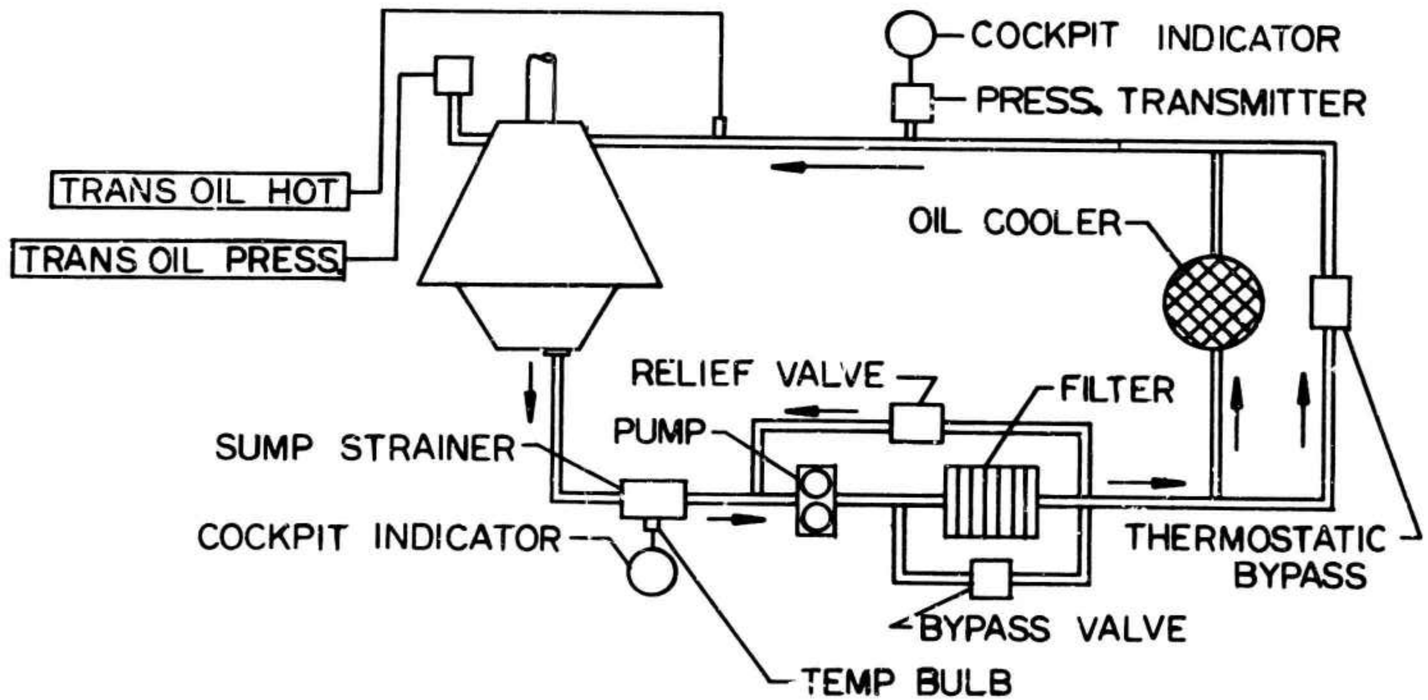


Figure 5. CH-54A Lubrication System Schematic.

diameter flexible line carries the cooled oil to a distribution manifold cast integrally as part of the main gearbox housing.

MISSION REQUIREMENTS

Typical mission power requirements for the CH-54A crane-type helicopter are presented in Table I.

TABLE I. COMPOSITE MISSION SPECTRUM - CH-54A AIRCRAFT

Flight Regime	Time (%)	SHP	MR (HP)	TR (HP)
Warm-up & Takeoff	2.20	5800	5136	464
	.62	2700	2284	216
Climb at 60 Knots	6.88	5300	4888	212
Hover (Steady State)	2.80	5150	4538	412
	10.54	4870	4280	390
	7.74	4700	4124	376
Hover (Maneuvers)	.05	6600	5435	965
	.10	6450	5305	945
	.09	6435	5470	765
	.10	6235	5135	900
	.26	6140	5190	750
	.14	5980	5250	530
	.20	5910	5010	700
	.79	5700	4970	530
	.53	5530	4800	530
	.62	2700	2284	216
Cruise at 85 Knots	30.94	3420	3083	137
	30.94	3280	2949	131
Partial Power Descent to Sea Level at 500 Ft/Min	4.46	2900	2630	70
	100.00	4080 (Prorate)	3800 (Prorate)	295 (Prorate)

DESIGN DATA AND POWER SPECTRUM

The design data upon which the transmission oil heat rejection investigation will be made is presented in Table II.

TABLE II. OIL HEAT REJECTION DESIGN DATA - CH-54A MAIN TRANSMISSION

Engine Data	
Engine	JFTD-12A-4A
Takeoff Power (HP)	6600
Output Speed (rpm)	9000
Main Gearbox Data	
Overall Ratio	48.6:1
Input Power (HP)	6600
Input Speed (rpm)	9000
Main Rotor Max Power (HP)	6000
Main Rotor Speed (rpm)	185.4
Tail Rotor Takeoff Max Power (HP)	1220
Tail Rotor Takeoff Speed (rpm)	3500
Oil Flow (gpm)	17.5
Power Loss (%)	2.2

ANALYSIS OF THE PROBLEM

OBJECTIVES OF STUDY

A number of research and development efforts have been and are being conducted to reduce the vulnerability of military helicopters. Among these are programs to provide strategically placed armor plating, self-sealing compounds/coatings for oil systems, and the use of dry lubricants. Many of these items show promise in providing additional protection for a number of critical helicopter components. In addition to these efforts, development tests are currently being performed on lubrication oil bypass systems to provide additional protection to the main drive train components by automatically bypassing the oil cooler in the event of a direct hit in the cooler core or connecting lines. In another program, a secondary, low-capacity system is being developed to provide lubricating oil to critical areas in the event of a malfunction of the primary lubrication system through projectile damage or pump malfunction. Helicopter main transmissions do not lend themselves to protection by means of armor plating, sealing compounds, etc., because of their size and complexity. Serious weight and performance problems result when these protective means are used. The oil bypass or secondary lubrication system provides for emergency operation only.

It is the objective of this study to provide methods of lubrication which do not require remote external cooling systems. Both long-range solutions for incorporation in future helicopter designs and more expedient means which may be retrofitted into existing systems will be investigated.

DESIGN REQUIREMENTS

The current state-of-the-art design practice for large helicopter power transmission systems is to dissipate at least 80 percent of the main gearbox power losses by means of an externally mounted oil-to-air heat exchanger. While other aircraft might take advantage of ram air for cooling, the helicopter power train cooling requirements are critical in hover, where near-maximum gearbox power losses must be dissipated at zero air velocity.

The most critical helicopter transmission cooling requirement usually occurs during a hot-day hovering condition where little or no ram air is available. The CH-54A main gearbox transmits as much as 5400 HP in a hover condition, as shown in Table I. The efficiency of this gearbox was established empirically and agrees quite favorably with the predicted (calculated) efficiency. The empirical data were obtained during the no-load lubrication test and initial power runs on the CH-54A Dynamic Systems

Test Stand. A curve of the power losses versus power transmitted is presented in Figure 6. When these losses are converted to heat,

$$\dot{Q} = \frac{33000 \text{ FHP}}{778} \quad (1)$$

$$= \frac{(33000)(120)}{778}$$

$$= 5100 \frac{\text{Btu}}{\text{min}}$$

By comparing the results of tests conducted on a fully insulated gearbox with the production configuration, it was established that 17 percent of the total heat load, or 867 Btu/min, is rejected through the housings by free convection and radiation. The balance, or 4240 Btu/min, must be dissipated through the heat exchanger. A summary of these and other CH-54A main gearbox cooling system data is presented in Table III.

TABLE III. DESIGN REQUIREMENTS - CH-54A COOLING SYSTEM

Parameter	Production System	Design Limitation
Maximum Input Power (HP)	6600	6600
Oil Flow Rate Through Distribution System (gpm)	17	14 to 17
Oil Flow Rate Through Cooler (gpm)	17	30
Oil Circuit Pressure (psi)	55	105
Oil Cooler Steady-State Heat Load (Btu/min)	4240	3905
Oil Cooler Heat Transfer Area, A_{oil} (ft^2)	43	-
Static Oil Sump Capacity (gal)	12	8
Oil Temperature out of Gearbox into Cooler (^oF)	245	293
Oil Temperature out of Cooler Into Gearbox (^oF)	195	250
Oil-Side Pressure Drop Across Cooler (psi)	8	50

TABLE III - Continued		
Parameter	Production System	Design Limitation
Air Temperature Into Oil Cooler, max (^{o}F)	104	104
Air Temperature out of Cooler (^{o}F)	160	-
Airflow Rate Through Cooler (cfm)	6900	-
Air-Side Pressure Drop Through Core (in. of H_2O)	8	-
Blower Shaft Power (HP)	12	-
Oil Cooler Dry Weight Less Support Structure (lb)	44	-
Blower Assy, Transition, & Blower Drive Weight (lb)	31	-

TRADE-OFF STUDY

After extensive literature research and resulting follow-up investigations, six oil heat rejection concepts were considered to be sufficiently promising to warrant further examination. These heat rejection concepts with potential for integration within a helicopter main transmission are as follows:

- A thermoelectric cooling system mounted within the transmission sump.
- A vapor cycle refrigeration system operating between hot gearbox oil and ambient air.
- Radiation and free and forced convection of heat by means of extended (finned) transmission case.
- Oil-to-air heat exchanger mounted within main rotor shaft.
- Oil-to-air heat exchanger integrated within the gearbox oil sump.
- Air cycle or vapor cycle "air-conditioning" system precooling ambient air at the inlet to an oil-to-air heat exchanger.

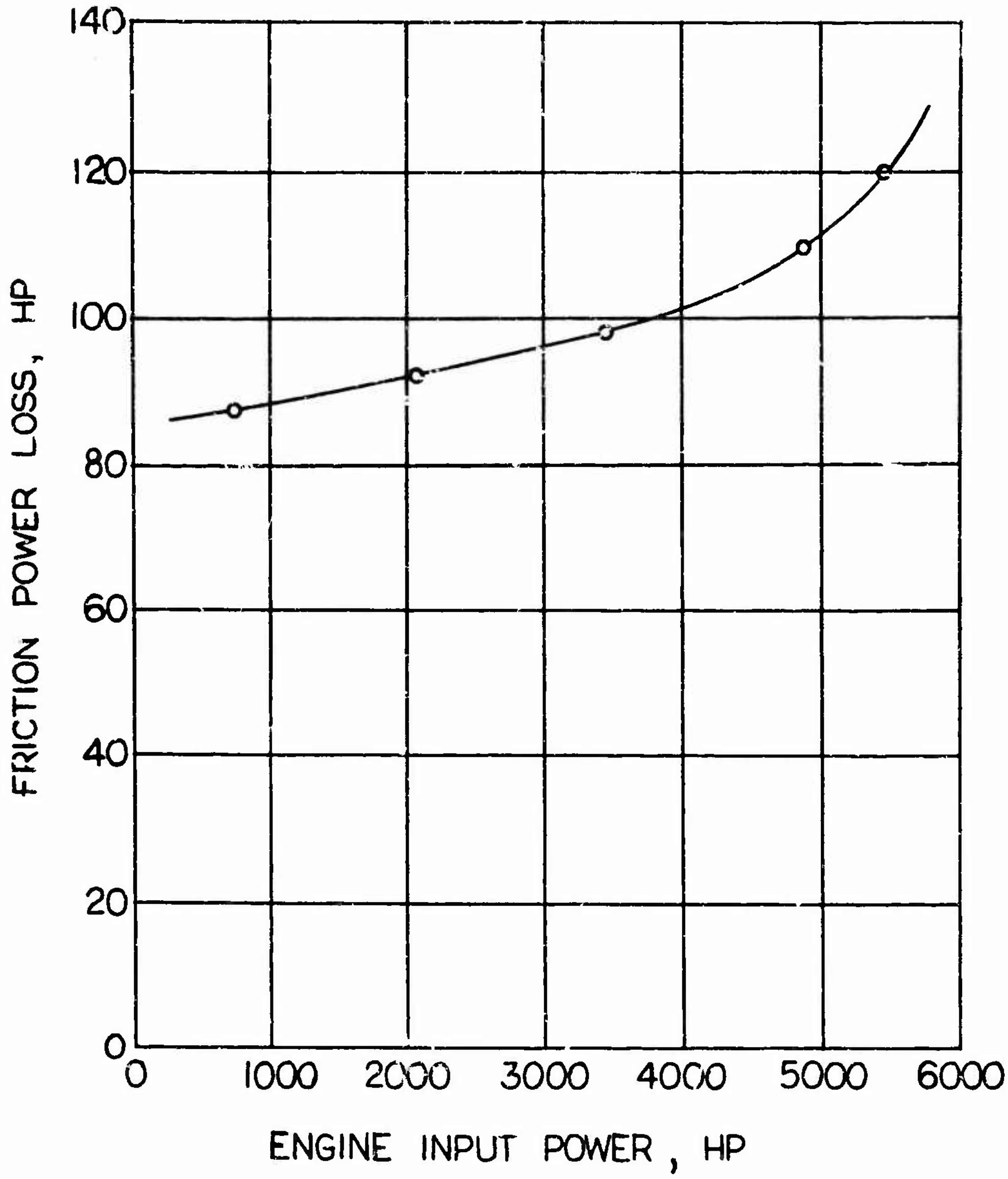


Figure 6. CH-54A Friction Horsepower Versus Engine Input Horsepower.

Thermoelectric Cooling

The thermoelectric cooling concept was investigated to determine its practicality for cooling large helicopter transmissions such as the CH-54A main gearbox. This system utilizes a thermoelectric heat pump consisting basically of a thermoelectric cold plate located in the gearbox sump and and a thermoelectric hot plate (heat sink) outside the box. Cooling occurs at the cold plate when large amounts of direct current are passed through the semiconductor junction. The heat is then rejected at the hot plate. A schematic of the system is shown in Figure 7.

At first glance, this system appears to be very attractive for transmission cooling because its primary cooling components are relatively compact, providing a rather small projected area. However, at the current state of the art, a typical thermoelectric cooler requires 5.5 watts of electrical power per junction to remove .087 Btu/min. Therefore, a considerable amount of electrical power (247 kw) and a number of thermoelectric junctions are necessary to meet the CH-54A main gearbox requirements of 3905 Btu/min.

The secondary power source required to supply the electrical power becomes very large and heavy, thus working at cross principles with the study goals. The thermal efficiency or coefficient of performance for thermoelectric cooling is approximately the same as that for commercial vapor refrigeration cooling. Until the thermoelectric performance for semiconductor materials increases, the thermoelectric heat pump is not practical for aircraft dynamic component cooling. Presently, commercial thermoelectric coolers of 4000-Btu/hr capability are available. An optimistic projection of the development of thermoelectric systems indicates that with improvements in materials and in the state of the art, coolers of 4000 Btu/min capability may be practical in the late 1970's. It is anticipated that thermoelectric cooling systems will always require more weight and more input power than conventional cooling systems.

Vapor Cycle Cooling

A vapor cycle refrigeration system operating in an oil-to-refrigerant-to-air sequence was evaluated as a possible transmission oil heat rejection concept. In this scheme, the refrigeration cycle operates between the transmission oil and ambient air, using an intermediate heat transfer medium. The vapor cycle system evaluated consists of the following components:

- Oil-to-refrigerant heat exchanger (evaporator) mounted integrally within the oil sump or main rotor shaft for protection.

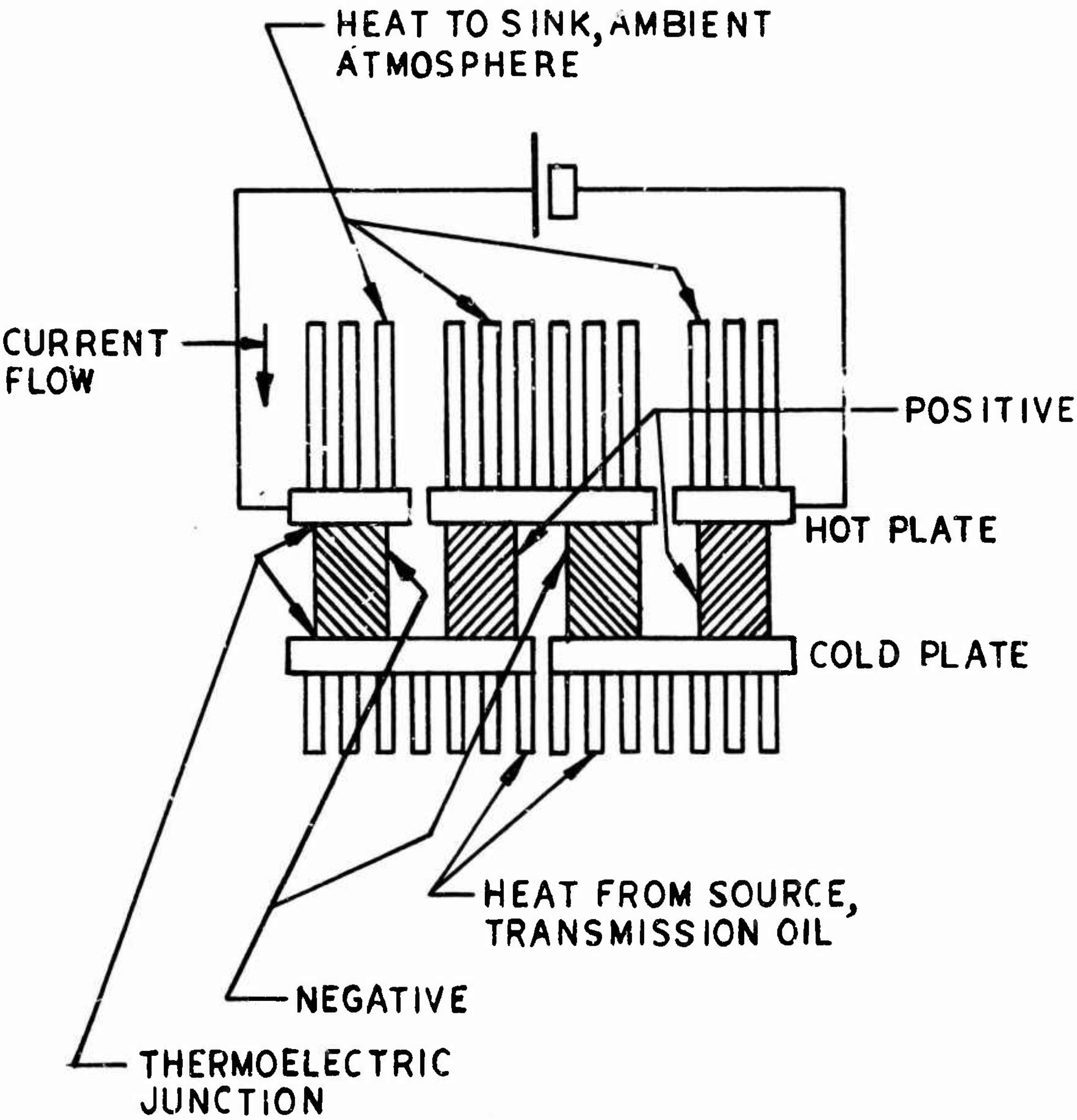


Figure 7. Thermoelectric Heat Pump.

- Shaft-driven compressor mounted on the accessory section of the main gearbox provided with armor protection.

- Externally mounted refrigerant-to-air heat exchanger (condenser) with appropriate transition ducting and condenser blower shaft driven from the main gearbox, all provided with armor protection.

- Refrigerant receiver to insure adequate refrigerant supply, externally mounted and armor protected.

- Necessary system plumbing and accessories required to complete the cycle, such as the expansion valve assembly and a suitable filter-drier assembly, externally mounted and armor protected.

A preliminary analysis indicates the vapor cycle-oil cooling system to have the following characteristics:

Power Requirements = 55 HP

Weight = 196 lb

On the basis of the initial analysis and the following observations, the vapor cycle-oil cooling system was judged to warrant further evaluation and is included in the Preliminary Design section of this report:

- A small-arms projectile hit sustained by the refrigeration system does not result in loss of gearbox oil. Therefore, the transmission can be operated for approximately 9 minutes without exceeding the "red line", providing sufficient emergency time to leave the combat area.

- It may be possible to obtain a greater heat transfer efficiency for the vapor cycle system than for a conventional oil-to-air heat exchanger if the upper temperature of the cycle can be made higher than the high oil temperature.

Finned Gear Case

There is some temperature at which the oil in a helicopter main transmission will stabilize if external oil cooling is not provided. For the current CH-54A configuration, this temperature is calculated to be approximately 500°F. At this condition, all heat rejection occurs through the housings and external surfaces by radiation and free convection (no forced convection

in the hover). If this temperature could be made to approach the 300°F gearbox operating limit, the oil cooling system as such could be eliminated, greatly reducing the projected area and gearbox vulnerability.

An investigation was made to determine if a sufficient heat transfer area could be developed to reject the full 5100 Btu/min. The additional surface area would be added by bonding or welding radial sheet-metal fins to the housing exterior walls. This approach has the additional advantage of providing good protection against small-arms projectiles for all but a trajectory parallel to the fins. A schematic of the system is shown in Figure 8.

To evaluate this approach, it is necessary to determine the percentage of heat dissipated through the existing housings by free convection and radiation and to determine the film heat transfer coefficient for convection from the housing to air for the unfinned case.

Radiation Losses

The radiation from a radiating surface may be expressed as

$$\dot{Q}_r = \Sigma \epsilon \sigma \; A_r (T_s^4 - T_a^4) \qquad (2)$$

For the purpose of this analysis, the main gearbox housings are broken down into simple geometric shapes, as summarized in Table IV, as are the results of equation (2). The total radiative losses are calculated to be 151.3 Btu/min from Table IV.

Convection Losses

In the hover mode, the most critical heat transfer condition, there is no significant airflow over the outside surfaces of the gearbox because the main gearbox is in the center, or eye, of the downwash of the main rotor. Therefore, little or no heat transfer takes place from forced convection from the housings. All housing and plumbing losses from the CH-54A, other than radiation, take place in the form of free convection. Therefore, the convection heat losses equal the total measured housing losses less radiation losses.

$$\dot{Q}_c = \dot{Q}_t - \dot{Q}_r \qquad (3)$$

$$= 867 - 151.3$$

$$= 715.7 \frac{\text{Btu}}{\text{min}}$$

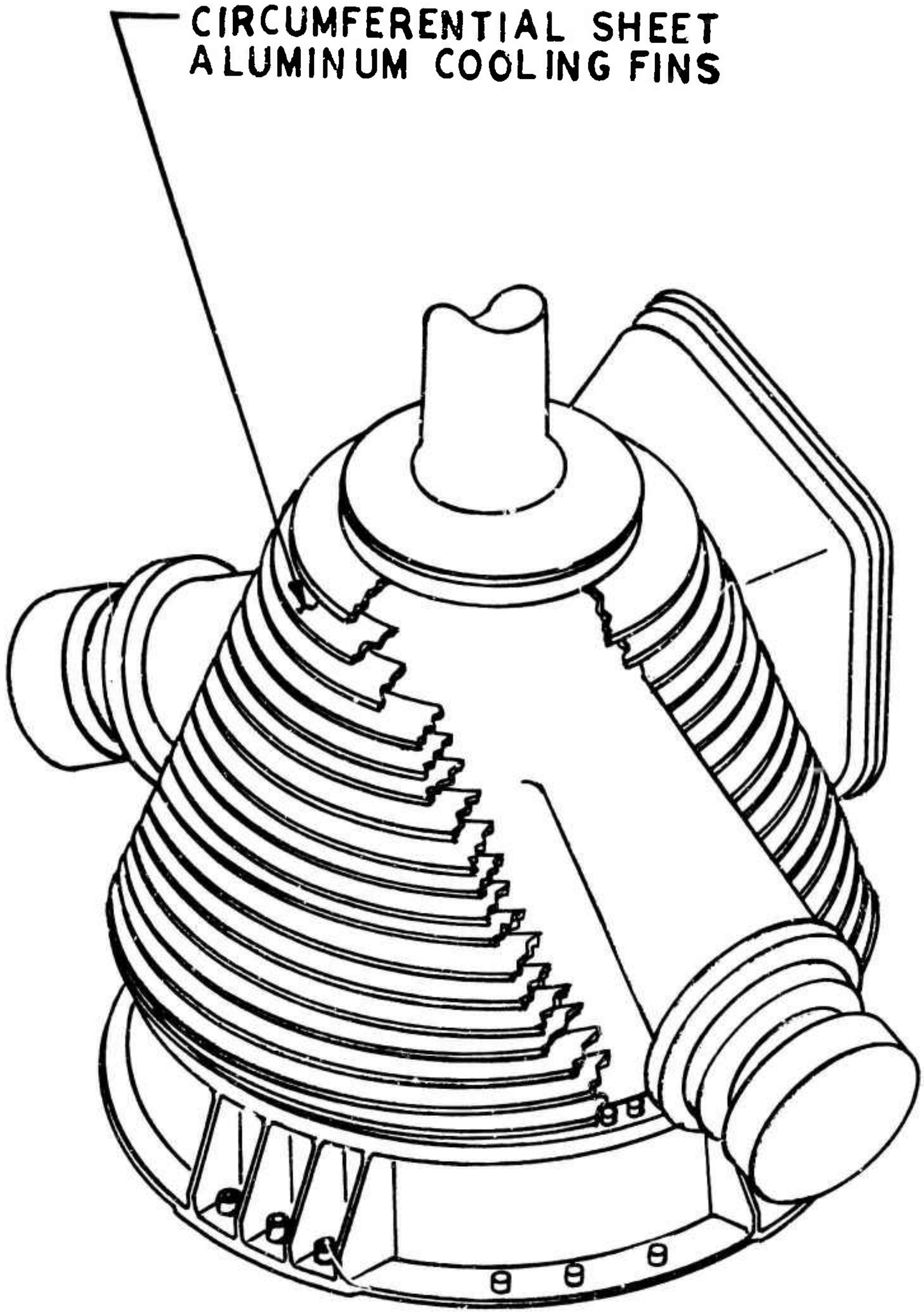


Figure 8. CH-54A Finned Main Housing.

TABLE IV. RADIATION AND CONVECTION LOSSES - CH-54A MAIN GEARBOX OPERATING AT 195°F

Nomenclature	Total Radiative Surface Area* (ft^2)	Total Convective Surface Area* (ft^2)	Mean-Skin Temperature** (°F)	Emissivity***	Radiative Losses (Btu/min)	Total Convective Losses (Btu/min)
Main Conical Housing Less Sump	19.7	19.7	200	0.60	34.3	-
Rear Housing Boss and Rear Cover	9.0	8.0	200	0.60	13.9	-
Engine Input Housing (2)	18.4	18.4	200	0.60	32.0	-
Bottom of Gearbox and Sump	12.6	-	245	0.60	34.4	-
Oil Drain Lines From Engine Inputs (2)	4.2	4.2	245	0.44	8.3	-
Oil Line From Cooler to Manifold	0.9	0.9	195	0.44	1.1	-
Oil Line From Gearbox to Cooler	1.6	1.6	245	0.44	3.2	-
All External Lubrication Lines	2.6	2.6	195	0.44	3.3	-
Oil Cooler Radiator	5.2	-	245	0.96	20.8	-
Total	74.2	55.4	-	-	151.3	715.7

*The bottom portion of the gearbox and the sump are not considered to be good convectors because free convection cannot occur in the stagnant hoist-well area. Radiation, however, will occur to other bodies such as the airframe, hoist, etc. The oil cooler itself is not considered to be a free convector, because all significant heat transfer occurs through forced convection induced by the blower; this is treated under oil heat rejection in the Design Analysis section of this report.

**Actual skin temperatures from CH-54A power transmission test-bed test results.

***Emissivity values: All housings and covers are magnesium alloy painted with olive-drab flat matte lacquer. All plumbing is bright stainless steel, unfinished. The radiator is brass painted with black flat matte lacquer.

For complex heat transfer situations, such as the CH-54A main gearbox exterior surfaces, the convection film heat transfer coefficient must be determined experimentally. Using the results of the CH-54A gearbox efficiency test and the analytical determination of radiation losses above, the coefficient is calculated from the following relation, where values of A_c and T_s are taken from Table IV.

$$\dot{Q}_c = \Sigma\, \bar{h}\, A_c\, (T_s - T_a) \tag{4}$$

$$= \bar{h}(49.6)(200 - 85) + \bar{h}(5.8)(245 - 85)$$

Solving equations (3) and (4) for $\bar{h}$,

$$\bar{h} = 6.47 \frac{\text{Btu}}{\text{hr ft}^2\ {}^\circ\text{F}}$$

These data are used in an analysis of a finned gearbox housing. The finned housing is the sole heat rejection medium. Table V summarizes this approach, including an evaluation of fin densities and air velocity at a temperature differential (ΔT) of 124°F (228°F housings; 104°F air).

Results of the analysis indicate that unless forced air convection can be provided over the finned surfaces, the required heat transfer can not be achieved. Providing a suitable blower, blower drive, ducting, and shrouding to force air over the fins adds complexity and additional vulnerability to the system. The use of finned gear housings, however, should be considered in the advanced design phase of any new program for the gear train of a combat helicopter to add protection and to help unload the oil cooling circuit. From the results tabulated in Table V, it becomes apparent that the blower requirements and weight penalties of the finned configuration capable of rejecting 5100 Btu/min of heat are prohibitive. The second fin density of Table V is estimated to weigh over 180 pounds using aluminum fins.

Main Rotor Shaft Heat Exchanger

A counterflow shell and tube-type cylindrical oil-to-air heat exchanger mounted integrally within the main rotor shaft of the main transmission has been investigated. The rotor shaft provides an impenetrable shield for the heat exchanger for maximum security from small-arms fire. The exchanger consists of a large number of small-diameter air tubes through which ambient air is drawn from the area above the main rotor. The shell side of the exchanger contains hot transmission oil circulated over the air tubes in one well-baffled pass.

TABLE V. HEAT TRANSFER SUMMARY - CH-54A FINNED AND UNFINNED MAIN GEARBOX

Fin Geometry and Density	Air Velocity (ft/sec)	Air-flow Rate (cfm)	Effectiveness (%)	Film Heat Transfer Coefficient (Btu/hr ft^2 °F)	$\dot{Q}$ Unfinned (Btu/min)	$\dot{Q}$ Finned (Btu/min)
Seventy-two .030-inch-thick, 2-inch-wide aluminum fins circumferentially spaced around main housing on .50 pitch. Develop total fin area of 432 ft^2.	0	0	85	1.1	875	1300
	20	3600	80	1.7	880	2000
	50	9000	50	8.0	1050	3700
Ninety .030-inch-thick, 2-inch-wide aluminum fins circumferentially spaced around main housing on .40 pitch. Develop total fin area of 527 ft^2.	0	0	80	1.1	875	1440
	20	3600	75	1.7	880	2180
	50	9000	40	8.0	1050	3900

Preliminary analysis of this design indicates that the inside diameter of the current production CH-54A rotor shaft significantly restricts the size of the heat exchanger as well as the airflow through the exchanger. These limitations make this design impractical for the present rotor shaft. However, an Engineering Change Proposal recommending the incorporation of the CH-53A rotor head and larger diameter rotor shaft is currently under consideration by the U.S. Army. Since the prospects for the approval of this ECP are good, an evaluation of this concept for the larger shaft was conducted, and the concept was found to be feasible.

The exchanger is mounted rigidly to the gearbox and does not rotate with the shaft. The appropriate clearance and upper support bearing for the exchanger are provided.

The blower operating in the draw-through mode is mounted to an integral oil sump transition duct assembly. The blower is shaft driven from the second-stage planetary output cage of the gearbox via short drive shaft couplings. The blower and transition duct portion of the system are located in the cargo hoist bay area of the CH-54A aircraft. Hot air is exhausted downward beneath the aircraft. Full analysis and layout drawings of this invulnerable system are included in the Preliminary Design section of this report.

Finned-Plate Sump Heat Exchanger

A crossflow finned-plate compact heat exchanger mounted integrally to a modified CH-54A main oil sump has been investigated. The heat exchanger consists of alternate air and oil passages between thin flat plates, through which ambient cooling air and hot transmission oil are passed. The air-side passages contain zig-zag interfinning. The oil passages terminate in appropriate oil-side headers containing the required plumbing provisions. A pusher-type shaft-driven blower supported by sump structure and inlet transition ducting is provided. Low-loss discharge ducting is also provided. A direct mechanical gear train from the second-stage planetary output cage drives the blower via short drive shaft and couplings.

The entire package is contained within the CH-54A cargo hoist bay for protection without interfering with the existing hoist or its components. Some armor plating may be provided to do double duty, protecting the cooling system as well as the sump. Modification to only the oil sump and lower housing of the main transmission is required to incorporate this concept. Airframe and hoist modifications are not required. Full analysis and layout drawings are included in the Preliminary Design section of this report.

Precooled Inlet Air - Air-to-Oil Heat Exchanger

During the analysis of the integral oil-to-air heat exchanger arrangements, it became apparent that a system to cool the inlet air to the heat exchanger might prove to be advantageous. An inlet air temperature below ambient provides a higher temperature differential and reduces the size of the heat exchanger. In the evaluation of this concept, "air-conditioning" by vapor cycle refrigeration and air cycle cooling by means of turbine expansion were considered. While these systems must be located external to the main gearbox, only partial oil cooling capability would be lost if they were damaged by small-arms fire.

The air-conditioning equipment required includes a refrigerant evaporator (heat exchanger) mounted i the inlet airstream to the integral oil-to-air exchanger. Also needed are components similar to those required for the vapor cycle-oil cooling system (though lower in capacity). For the air cycle system, a gearbox-driven compressor-turbine, which discharges expanded subcooled air into the inlet airstream of the integral gearbox oil cooler, is required. This approach combines the component requirements and complexity of both the integral heat exchanger and the refrigerant systems. On this basis, the reliability and vulnerability of this combined system was judged to be less favorable than the other systems evaluated.

Summary

As a result of this initial trade-off study, including preliminary analyses of each transmission cooling system configuration, three systems were selected for further evaluation in the preliminary design phase. These systems, which were considered to best meet the vulnerability, reliability, and performance objectives, are:

- Main rotor shaft cylindrical heat exchanger
- Integral oil sump heat exchanger
- Vapor cycle refrigeration system

Based on the analyses, all of the concepts studied in this phase of the program, with the exception of thermoelectric cooling, could be developed to yield the required heat transfer. The three systems that are carried through the preliminary design phase appear to be the most feasible solutions.

PRELIMINARY DESIGN

INTRODUCTION

On the basis of the initial trade-off study, three oil heat rejection systems capable of being integrated within the main gearbox have been selected for further evaluation. In this phase, preliminary (layout) design drawings have been made for each system. A heat transfer analysis for each system investigated, including efficiency, blower power requirement, and relative weights, has been performed. In addition, analyses establishing the comparative reliability, maintainability, and vulnerability have been made.

ROTOR SHAFT COOLING SYSTEM

System Description

In this system, a cylindrical heat exchanger is mounted within the main shaft of the CH-54A main gearbox. The cooler is supported on the lower bearing housing for the rotor shaft and is stationary. The inlet oil line is attached to the lower header; the outlet is attached to an integral line running to the upper header. A multiple-stage axial fan draws ambient air from above the main rotor through the heat exchanger and discharges it through ducting into the cargo hoist bay of the airframe. The system is shown in Figure 9.

Heat Exchanger

The proposed main rotor shaft oil-to-air heat exchanger is of the high-performance shell and tube design. For this concept, geometry dictates that the exchanger be round, that the airflow be axial through the long dimension, and that the frontal area be relatively small in comparison with exchanger heat transfer requirements. The tube bundle is the most efficient core configuration for a heat exchanger with these envelope dimensions.

The shell is type 321 stainless steel of .032 inch wall thickness with an outside diameter of 6.875 inches; it is approximately 20 inches long, including inlet and outlet oil headers. The shell contains 1680 air tubes closely packed in an equilateral staggered pattern, as shown in Figure 10. The air tubes have a slip fit with the headers and are swaged or dip-brazed to effect an oil-tight assembly. The oil-side baffling doubles as the tube intermediate support structure along the length of the exchanger to maintain tube spacing and straightness and to eliminate tube response and deflection when the exchanger is subjected to vibration. The tubes are fabricated from 321 stainless steel to ensure galvanic compatibility and to

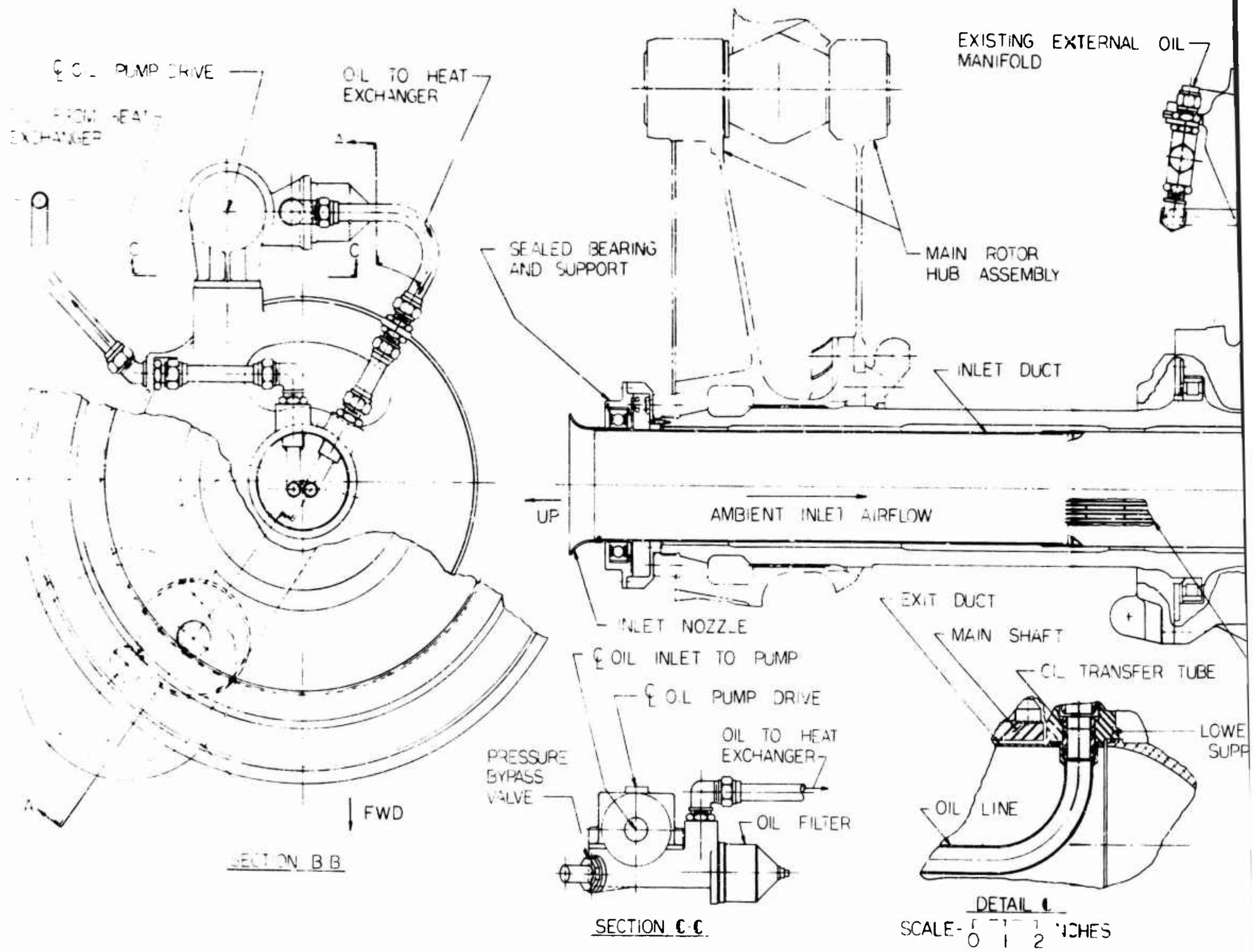
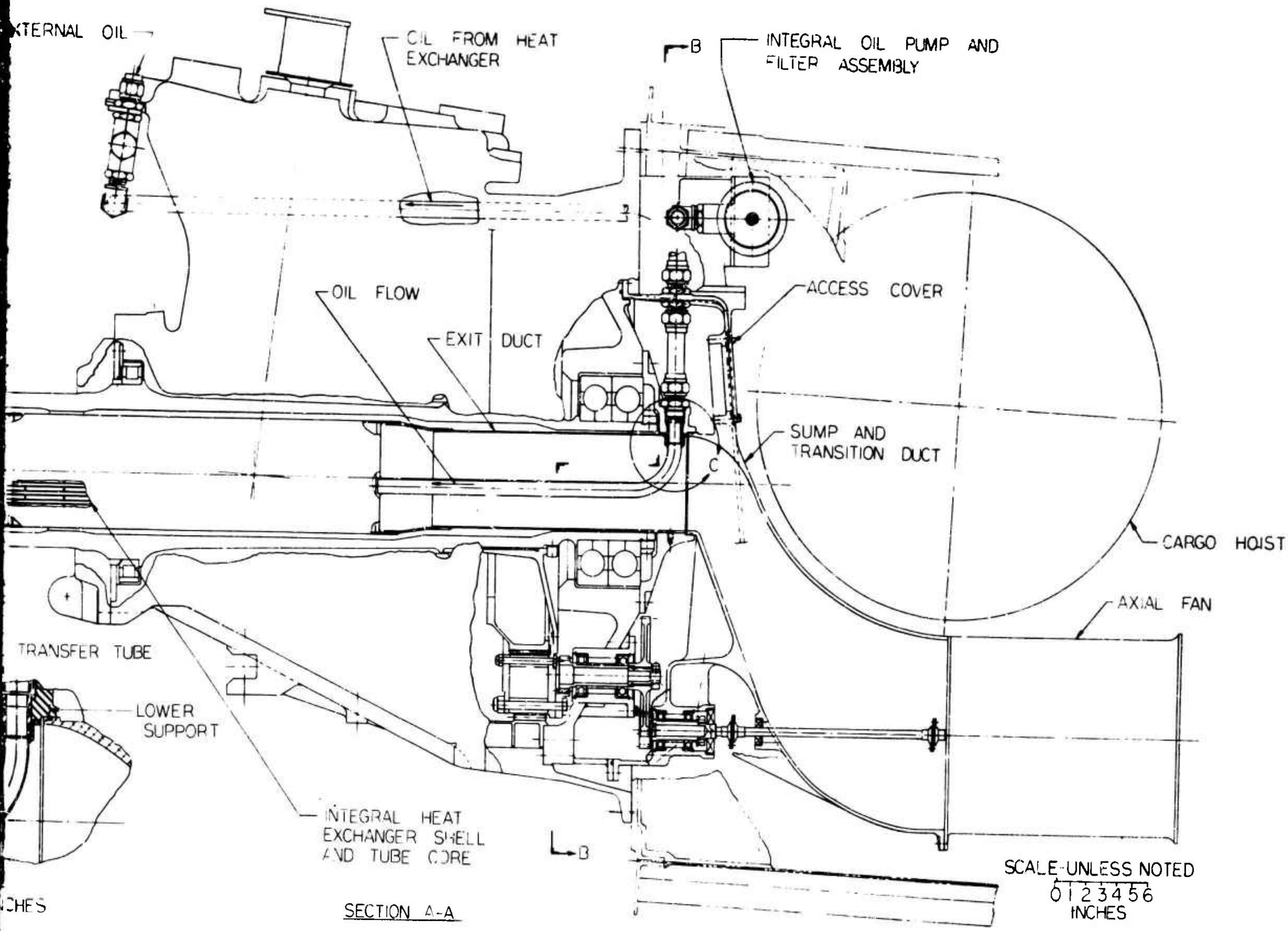


Figure 9. CH-54A Rotor Shaft Cooling System.



XTERNAL OIL
OIL FROM HEAT EXCHANGER
B
INTEGRAL OIL PUMP AND FILTER ASSEMBLY
OIL FLOW
EXIT DUCT
ACCESS COVER
SUMP AND TRANSITION DUCT
C
CARGO HOIST
AXIAL FAN
TRANSFER TUBE
LOWER SUPPORT
INTEGRAL HEAT EXCHANGER SHELL AND TUBE CORE
B
SCALE-UNLESS NOTED
0 1 2 3 4 5 6
INCHES
CHES

SECTION A-A

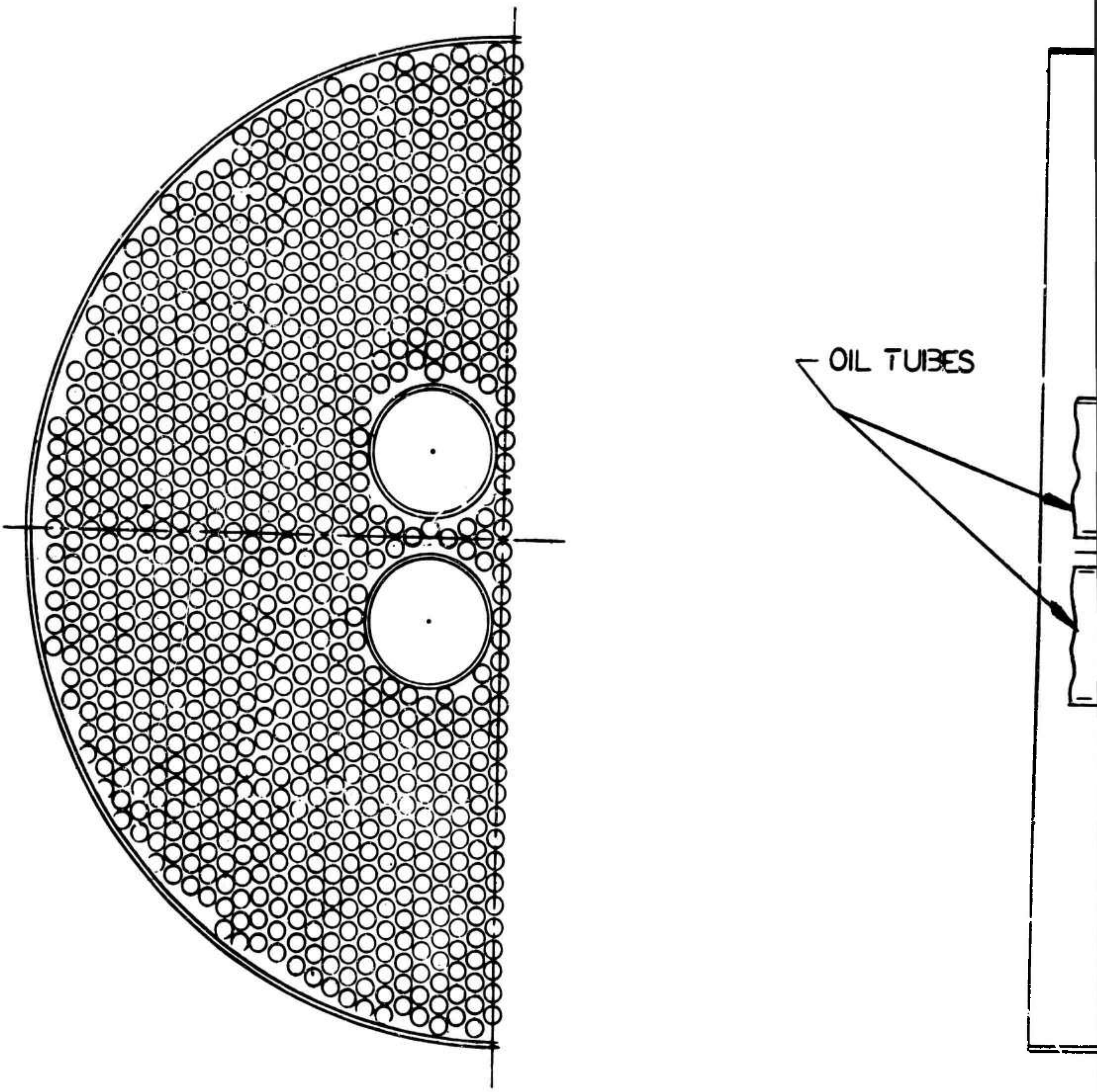
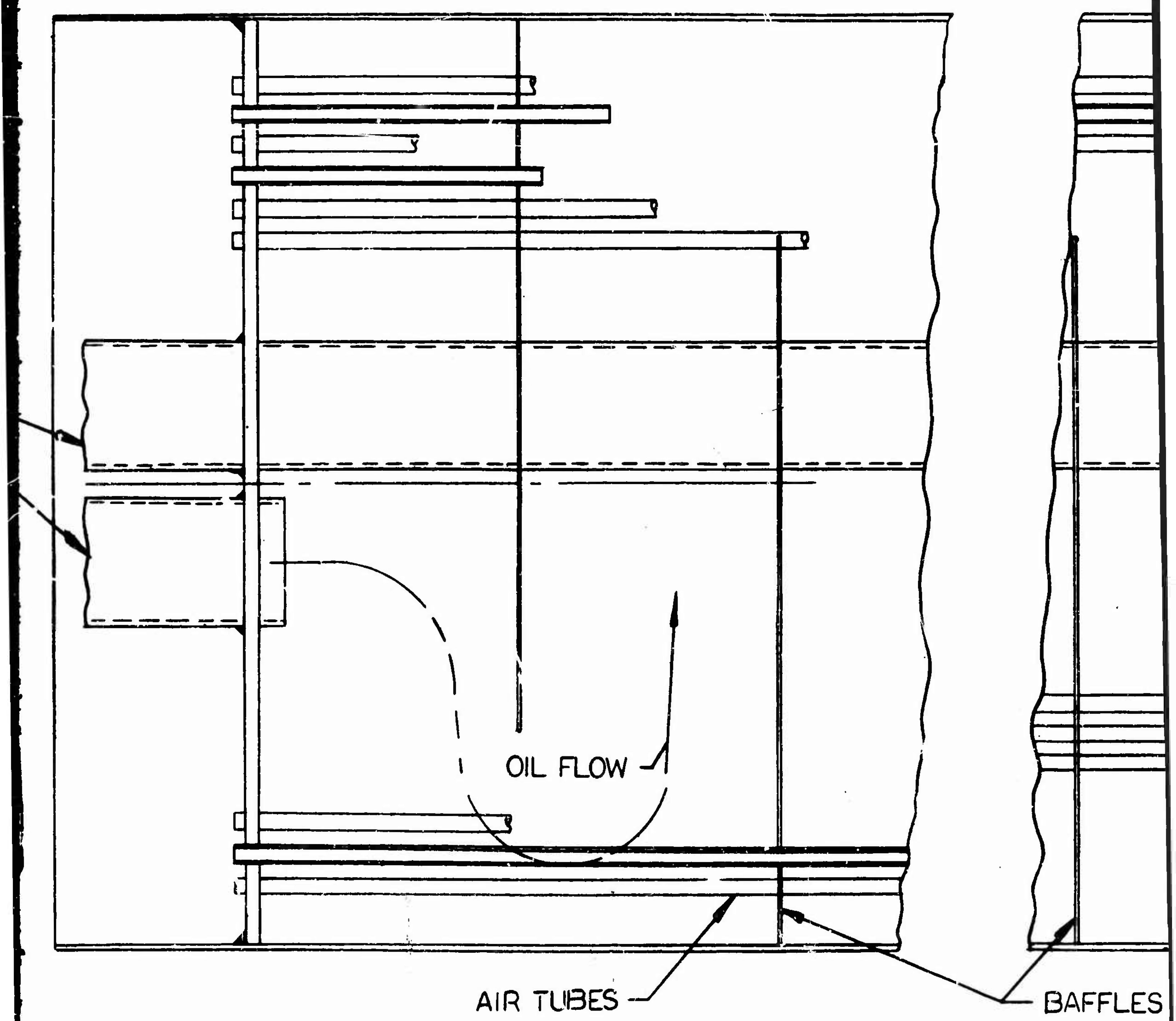
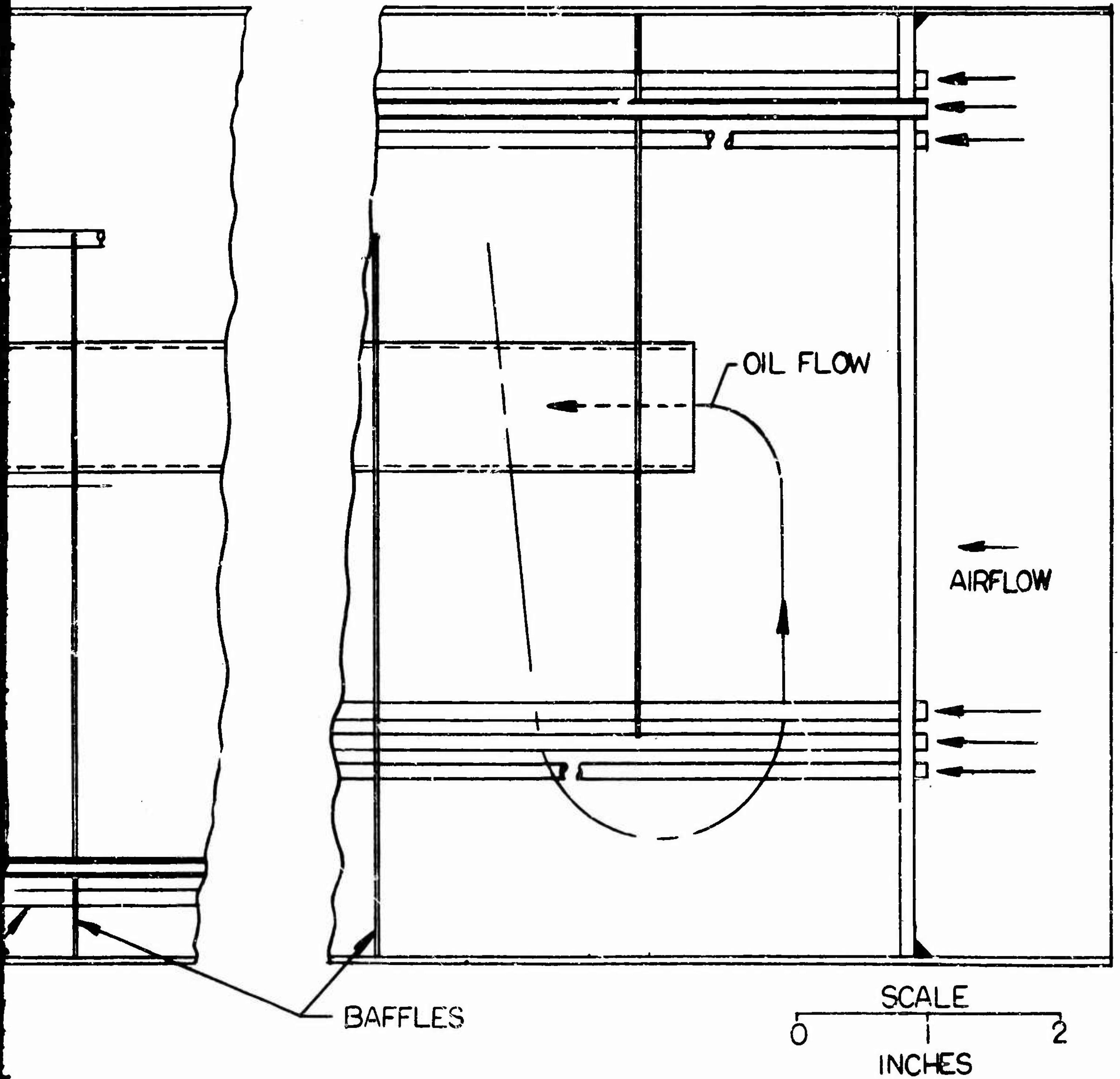


Figure 10. CH-54A Rotor Shaft Heat Exchanger.

A


OIL FLOW
AIR TUBES
BAFFLES


OIL FLOW
AIRFLOW
BAFFLES
SCALE
0
1
2
INCHES

eliminate the need for a floating header to compensate for differential thermal expansion within the exchanger. The tubes have a .125-inch outside diameter and a .005-inch wall thickness. The tubes are spaced with a .150-inch pitch (.150 inch between centers), providing an oil passage between tubes of .025 inch.

Note: According to some heat exchanger manufacturers, the smallest size air tubes used in most applications are of .250-inch outside diameter. Fabrication costs increase when smaller diameter tubes are used. Two manufacturers indicated, however, that .125-inch air tubes would not represent any fabrication hardship and that the cost would increase roughly in linear proportion to the increased number of tubes.

Multiple-Stage Axial Blower

The air is drawn through the tube bundle heat exchanger by a two-stage axial blower. The blower is shaft driven from the lower portion of the main gearbox, as shown in Figure 9. The blower contains two sets of rotor vanes designed to move air in the direction parallel to the axis of the rotors. Two sets of stator vanes serve to channel the flow of air and to reduce turbulence through the blower. The rotor and shaft assembly is supported by two or more sealed, prelubricated rolling element bearings mounted in the blower housing.

Blower Drive

The multiple-stage blower is shaft driven from the second-stage planetary output cage via a compact step-up gear train and a short length of drive shafting, as shown in Figure 9. An internal ring gear bolted to the planetary cage drives a gear and pinion on a countershaft, which in turn drives a second mesh to provide the final step-up to the drive shaft of 9000 rpm. The drive shaft is provided with multiple-disc flexible couplings at both ends to accommodate misalignment and deflection.

Ducting and Accessories

The transition from the bottom of the main shaft heat exchanger to the inlet of the blower is a structural portion of the oil sump. The sump and transition duct is a casting or a weldment. The oil sump and oil pump suction and mounting arrangement are identical to the current CH-54A configuration. The static oil capacity of the gearbox is the same as for the current production unit. The inlet ducting to the heat exchanger is extended the full length of the rotor shaft and is supported by a double-sealed, prelubricated light series ball bearing mounted in a bearing housing

supported in the rotor hub. The sealed bearing provides the necessary sealing to prevent dirt and water from entering the sump and internal portions of the gearbox. The gearbox operating temperature is regulated by a thermostatic bypass valve between the inlet and outlet oil lines of the heat exchanger mounted in a cored bulkhead of the sump and pump support. The thermostat incorporates a high-pressure bypass to permit sufficient oil flow for cold weather operation.

Lubrication System Operation

The oil lubrication system operation for the main rotor shaft cooler design shown in Figure 11 is as follows:

> Hot oil from the gearbox sump is drawn into the lubricating pump through a 100-micron oil screen. The oil is pumped through a 40-micron filter and the rotor shaft heat exchanger to the pressure regulator. The cooled oil flow is divided with approximately 40 percent going to the gearbox lubrication jets and 60 percent bypassing directly back into the oil sump, where it mixes with the hot oil returning from the extremities of the gearbox.

The existing CH-54A gearbox lubrication system differs from the integral system only in that the flow division in the lubrication system occurs before the cooler instead of after the cooler. One-third of the flow is bypassed directly back to the inlet side of the pump. Two-thirds is passed through the cooler and on to the distribution system. The relocation of the regulator is proposed for the shaft cooler system, since the analysis indicates that both the oil-side and the air-side performance are improved when the mass flow rate of oil is increased from the current CH-54A rate of 17 gpm to 30 gpm.

Performance Analysis

Introduction

The shell and tube heat exchanger contained within the main rotor shaft imposes a restriction on the amount of cooling air that can be passed through it because of the small frontal area. The heat transfer analysis is therefore critical on the air side. A preliminary calculation is performed with optimized air-side parameters to determine blower requirements.

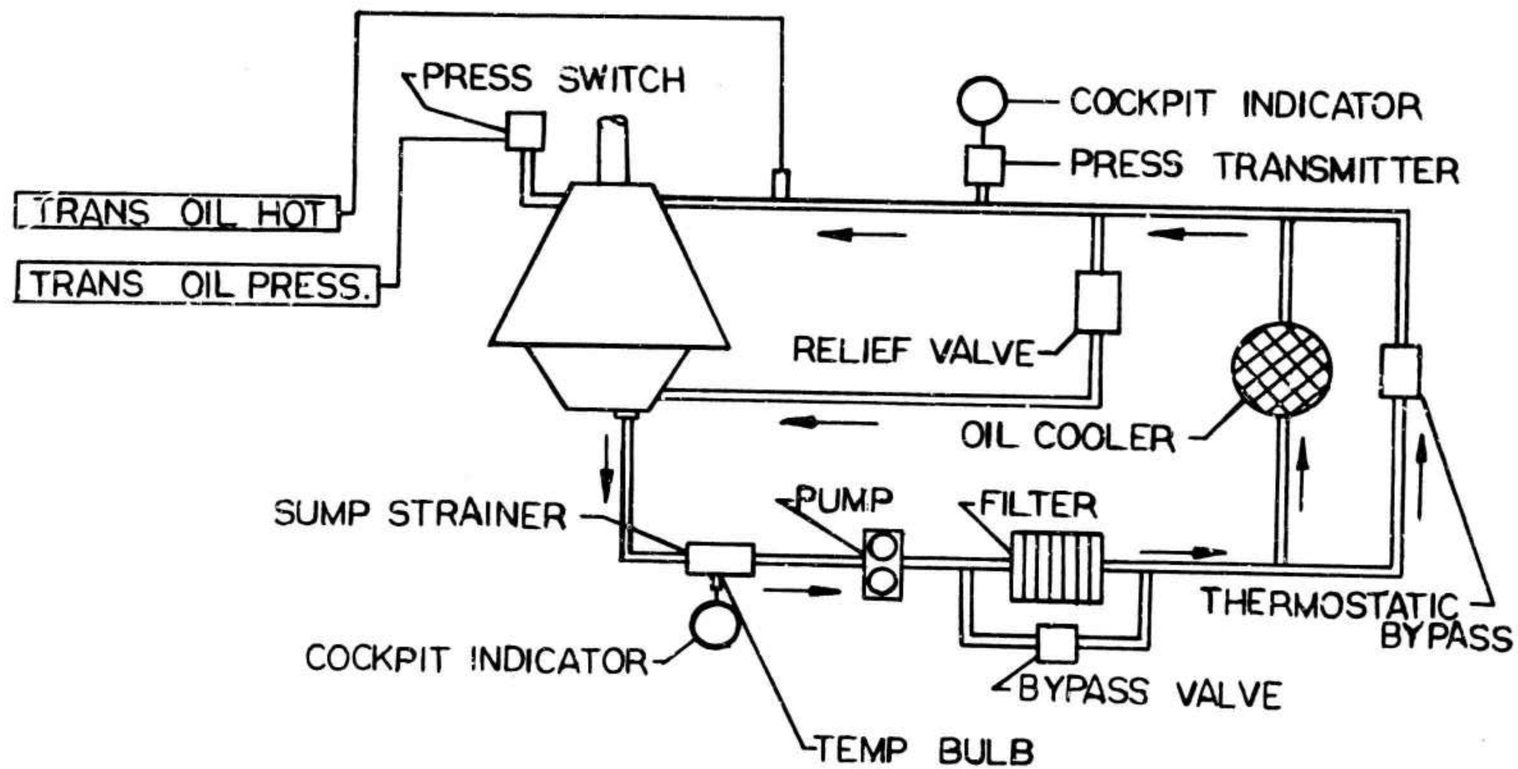


Figure 11. CH-54A Rotor Shaft Cooling System Lubrication Schematic.

Preliminary Analysis

Blower Requirements

The gearbox hot oil temperature in the heat exchanger is 245°F on the CH-54A. Hot-day ambient air temperature equals 104°F. For a perfect heat exchanger, the following relation would be true:

$$\Delta T_{max} = (245°F - 104°F) \tag{5}$$

$$= 141°F$$

However, the optimum counterflow shell and tube heat exchanger effectiveness equals 0.9.

$$\Delta T_{act} = e \ \Delta T_{max} \tag{6}$$

$$= (.9)(141)$$

$$= 127°F$$

The mass flow rate of air is calculated as follows:

$$\dot{m}_a = \frac{\dot{Q}}{c_p \ \Delta T_{act}} \tag{7}$$

$$= \frac{4200}{(.241)(127)}$$

$$= 137 \ \frac{lb}{min}$$

The mass velocity of air is calculated as follows:

$$G_a = \frac{\dot{m}_a}{A_{cs}} \tag{8}$$

$$= \frac{(137)(144)(60)}{15.6}$$

$$\approx 76,000 \ \frac{lb}{hr \ ft^2}$$

The core friction loss through the heat exchanger is calculated from the following relation:

$$\Delta P_{core} = \left[\frac{G_a^{\,2}}{2g\ \rho_1}\right]\left[\frac{f_d\ A_a\ L\ \rho_m}{A_{cs}\ \rho_2}\right] \quad (9)$$

$$= \left[\frac{(76,000)^2}{(2)(4.17 \times 10^8)(.0705)}\right]$$

$$\left[\frac{(.0069)(47.4)(1.85)(144)(.054)}{(15.6)(.037)}\right]$$

$$= 800\ \frac{lb}{ft^2}$$

$$= 153 \text{ in. of } H_2O$$

The core loss through the heat exchanger due to flow acceleration from restriction in the air-side area is calculated from the following relation:

$$\Delta P_{flow\ acc} = \left[\frac{G_a^{\,2}}{2g\ \rho_1}\right]\left[2\left(\frac{\rho_1}{\rho_2} - 1\right)\right] \quad (10)$$

$$= \left[\frac{(76,000)^2}{(2)(4.17 \times 10^8)(.0705)}\right]\left[2\left(\frac{.0705}{.037} - 1\right)\right]$$

$$= 175\ \frac{lb}{ft^2}$$

$$= 34 \text{ in. of } H_2O$$

$$\Delta P_{total} = \Delta P_{core} + \Delta P_{flow\ acc} \quad (11)$$

$$= 153 + 34$$

$$= 187 \text{ in. of } H_2O$$

The blower air horsepower required to move air through the heat exchanger is calculated as follows:

$$AHP = \frac{(62.3)(\Delta P_{total})\ (\dot{m}_a)}{(12)\ (33,000)\ \rho_2} \qquad (12)$$

$$= \frac{(62.3)(187)(137)}{(12)(33,000)(.037)}$$

$$= 108\ HP$$

The blower shaft horsepower is calculated as follows:

$$SHP = \frac{AHP}{\eta}$$

$$= \frac{108}{.7}$$

$$= 154\ HP$$

Summary

When a multiple-stage axial vane blower is operating against a suction head of 187 inches of H_2O, or approximately 6.16 psi, it is doubtful that the blower would have an efficiency as high as 70 percent; but, assuming that it did, the shaft power required to drive the blower would be 154 HP. A blower of this capacity would be approximately 18 inches in diameter by 23 inches in length. It is not feasible to fit a blower of this envelope into the transmission/cargo hoist bay area of the CH-54A airframe.

Reduction of Gearbox Heat Load

Since the concept of protecting the oil circuit within the main rotor shaft provides a feasible approach to the integration of the cooling system within the gearbox to minimize vulnerability, further design iterations were made to improve air-side parameters.

The cooling requirements and efficiencies of the CH-54A and other helicopter main transmissions were reviewed. Several helicopter main transmission systems, including those of the SH-3A and CH-3C, have operated at main gearbox oil-out temperature levels of 293°F for many years.

Since these transmissions are similar to the CH-54A both in operation and in materials, the oil-out temperature of the CH-54A main transmission will be raised from 245° to 280°F. Correspondingly, oil temperature out

of the cooler is the maximum practical operating level consistent with good design practice for helicopter transmission systems for the following reasons:

1. The magnesium alloy cast housings employed exclusively on all gearboxes, all models, experience creep phenomena in areas subjected to load at housing temperatures in excess of 300°F.

2. The shafting and gearing manufactured from SAE 9310 steel alloy experience "drawing" of the carburizing or reduction in case hardness at temperatures in excess of 300°F.

3. For every bearing and seal installed in magnesium housings or supports, a press-fit steel liner is employed to control wear and differential thermal expansion. At 250°F actual liner-to-housing interface temperature, the press-fit in the housing goes to zero at the worst dimensional tolerance condition, due to differential thermal expansion between steel and magnesium. At gearbox operating temperatures in excess of 300°F, liner interface temperatures may approach 250°F, depending on the ability of the housing at that point to dissipate heat. This condition is avoided to eliminate loose or spinning liners that would introduce wear or eccentricity.

4. At oil temperatures above 300°F, the oil viscosity index may approach undesirably low values where load-carrying capacity and oil film strength will be inadequate. This condition will increase wear and will reduce the time between overhauls of the component.

Operating the gearbox at the higher temperature level will result in the following improvements that make the concept of an integral cooling system within the main rotor shaft fully feasible and practical:

1. The oil shearing portion of the friction horsepower of the gearbox decreases with decreased oil viscosity, thereby decreasing the overall heat load.

2. The percentage of losses from the housings through radiation and free convection increases with higher skin temperature, decreasing the amount of oil heat that must be rejected through the cooling system.

3. The oil heat rejection in the oil cooler is more efficient because of the higher temperature differential.

Reduction of Gearbox Friction Horsepower

Results of empirical programs conducted by rolling-element bearing manufacturers indicate that as oil inlet temperature to the bearing increases, the friction power generated by churning and shearing oil decreases because of decreasing oil viscosity. The dynamic portions of the CH-54A main gearbox are gears and bearings. The same reasoning and analysis developed through bearing research will be applied to both bearing friction and gear mesh friction calculations, as shown below.

The friction horsepower developed in an oil-lubricated rolling-element bearing decreases with decreasing oil viscosity, per Reference 1, when DN (bearing bore times rpm), total imposed load, and oil flow rate are held constant, in accordance with the following relations:

$$\text{FHP} = \frac{f\ R_s\ W\ N}{63,025} \tag{13}$$

$$\text{FHP} = K\ \mu^{0.25} \tag{14}$$

When equations (13) and (14) are combined, an expression for coefficient of friction varying with oil viscosity is obtained:

$$f = K^1\ \mu^{0.25} \tag{15}$$

From the plot of viscosity versus temperature for SATO 35 oil, Figure 12, the viscosity decreases from 20 lb/hr-ft to 12 lb/hr-ft when the temperature is raised from 195° to 246°F. This results in a decrease of 13 percent in the friction horsepower of each rolling-element bearing.

While there are no empirical data available to predict accurately the effects of oil viscosities and friction horsepower on gear mesh efficiencies, from Reference 2, it is believed that similar relationships for the coefficient of friction apply to gears as well as to bearings. When the equation for gear mesh efficiency of Reference 3 is used, a good approximation of efficiency can be obtained from the following relation, where the coefficient of friction varies with oil viscosity in accordance with equation (15):

$$\eta = 1 - \left[\frac{\cos\phi}{\cos\phi_n\ \cos\psi}\right]\left[\frac{1+\frac{1}{m}}{\beta_a + \beta_r\ \cos\psi}\right]\left[\frac{f}{2}\left(\beta_a^{\,2} + \beta_r^{\,2}\right)\right] \tag{16}$$

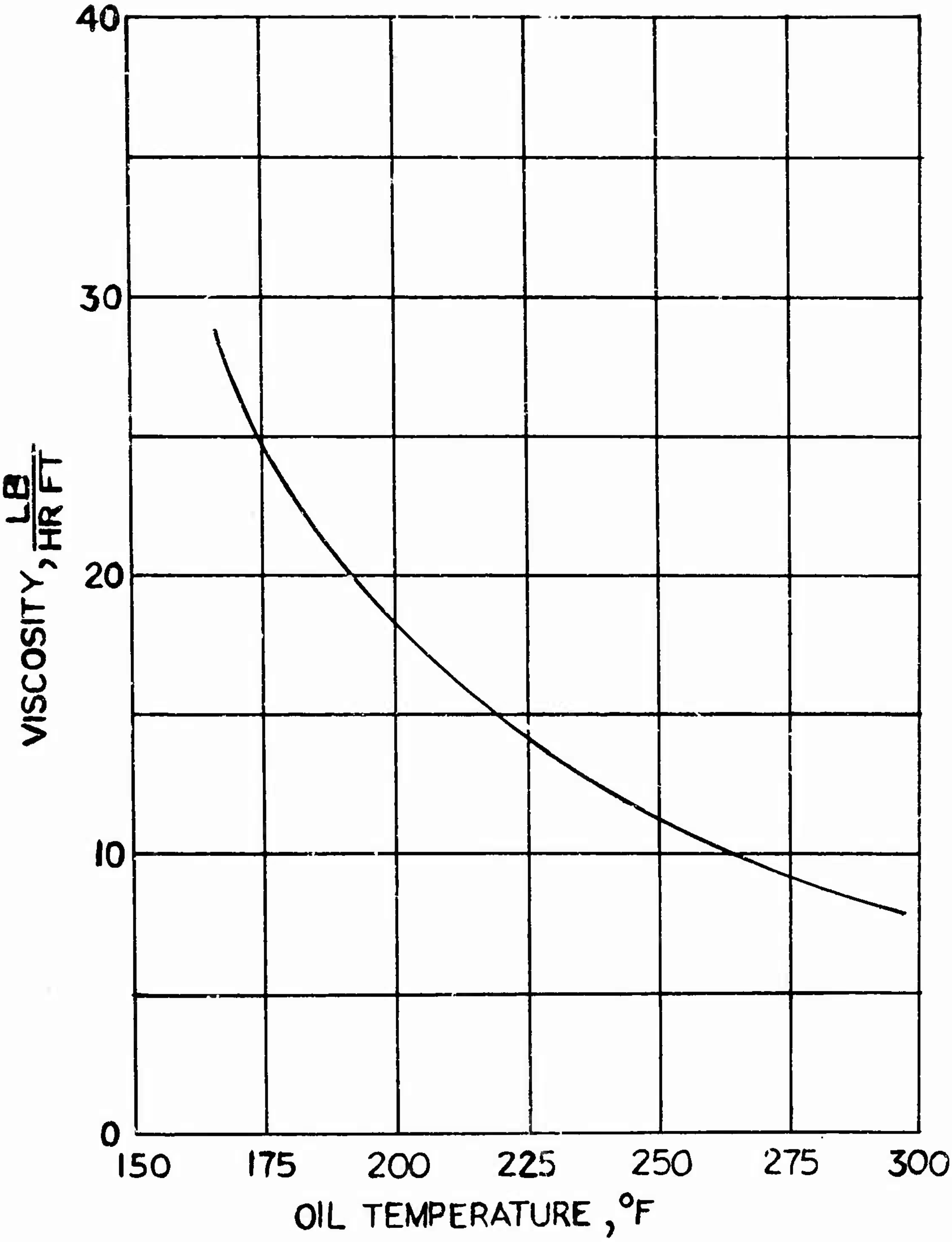


Figure 12. Oil Viscosity Versus Temperature.

When the operating temperature is raised from 195° to 246°F, a decrease in friction horsepower of 9 percent is experienced in each gear mesh.

A comparison of the predicted friction horsepower for the CH-54A main gearbox for oil temperatures of 195° and 246°F is presented in Table VI.

The total heat load of the gearbox operating at a higher temperature is calculated as follows (FHP = 113.5 from Table VI):

$$\dot{Q} = (113.5)(42.44) \tag{17}$$

$$= 4820 \text{ Btu/min}$$

TABLE VI. CALCULATED FRICTION HORSEPOWER - CH-54A MAIN GEARBOX

Gearbox Element	FHP 195°F	FHP 246°F
Gear Meshes	90.8	84.0
Rolling-Element Bearing, Primary Drive	9.6	8.4
Accessory Section Gears and Bearings	8.0	7.0
Windage and Seal Drag	10.1	10.1
Lubrication Pump	1.5	4.0*
Total	120.0	113.5

*4.0 HP required for lubrication system of 30 gpm at 55 psi.

Increased Heat Rejection Through Housings

Radiation Losses: When the gearbox operating temperature is increased from 195° to 246°F, the corresponding radiating skin surface

temperature will not follow degree for degree, because the skin temperature is diminished by the increasing rate of heat transfer from the housing walls through radiation at this higher skin temperature. Actual test data are required to determine the higher skin temperature, but the required testing is beyond the scope of this study. It is therefore assumed that the skin temperature will follow in proportion, as expressed in the following relation:

$$\frac{T_{intlow}}{T_{skinhigh}} = \frac{T_{inthigh}}{T_{skinhigh}} \qquad (18)$$

The radiating skin temperature for the main housing is calculated as follows:

$$T_{skinhigh} = \frac{(280)(200)}{245}$$

$$= 228^{\circ}F$$

Calculated radiation loss at the housing skin temperature of 228°F and an ambient air temperature of 104°F is 156 Btu/min.

Convection Losses: Using the value of the film heat transfer coefficient for the main housings to ambient air obtained from equation (4) of $\bar{h}$ = 6.47 Btu/hr-ft^2 °F, and the increased housing skin temperature of 228°F, the calculated convection loss is 759 Btu/min. The assumption that the value of $\bar{h}$ remains constant at the different housing skin temperature is valid for the small temperature differential experienced.

A summary of radiation and convection losses at the increased gearbox operating temperature of 246°F is presented as Table VII.

Total Oil System Heat Load

The main rotor shaft shell and tube heat exchanger and the other integral oil cooling systems have been designed to dissipate a total oil heat load as determined below:

$$\dot{Q} = \dot{Q}_t - \dot{Q}_r - \dot{Q}_c \qquad (19)$$

$$= 4820 - 156 - 759$$

$$= 3905 \ \frac{\text{Btu}}{\text{min}}$$

TABLE VII. RADIATION AND CONVECTION LOSSES - CH-54A MAIN GEARBOX OPERATING AT 246°F

Nomenclature	Total Radiative Surface Area* (ft^2)	Total Convection Surface Area* (ft^2)	Mean Skin Temperature** (°F)	Emissivity***	Radiative Losses (Btu/min)	Total Convective Losses (Btu/min)
Main Conical Housing Less Sump	19.7	19.7	228	0.6	41.6	266
Rear Housing Boss and Rear Cover	8.0	8.0	228	0.6	16.9	108
Engine Input Housing (2)	18.4	18.4	228	0.6	38.8	249
Bottom of Gearbox and Sump	12.0	-	280	0.6	41.0	-
Oil Drain Lines From Engine Inputs (2)	4.2	4.2	280	0.44	10.5	80
Oil Line From Cooler to Manifold	1.0	1.0	250	0.44	1.9	15
All External Lubrication Lines	2.6	2.6	250	0.44	5.3	41
Total	65.9	53.9	-	-	156.0	759

*The bottom portion of the gearbox and the sump are not considered to be good convectors because free convection cannot occur in the stagnant hoist-well area. Radiation, however, will occur to other bodies such as the airframe, hoist, etc. The oil cooler itself is not considered to be a free convector, because all significant heat transfer occurs through forced convection induced by the blower; this is treated under oil heat rejection in the Design Analysis section of this report.

**Actual skin temperatures from CH-54A power transmission test-bed test results.

***Emissivity values: All housings and covers are magnesium alloy painted with olive-drab flat matte lacquer. All plumbing is bright stainless steel, unfinished. The radiator is brass painted with black flat matte lacquer.

Design Analysis

A detailed analysis of the main shaft heat exchanger design is included as Appendix I. The calculated performance of this system is summarized in Table VIII.

Weight Analysis

The calculated weights for the component parts of the main shaft cooling system are summarized in Table XIX.

Discussion

While the design analysis of Appendix I is self explanatory, some additional comments/insight into the approach are offered in this section.

It became readily apparent early in the preliminary design phase of this approach that since the envelope available for the shaft cooler can not be increased without a major redesign of the gearbox, maximum effort had to be taken to optimize both the air-side and the oil-side performance. This is accomplished most significantly by increasing the oil-out temperature of the gearbox and by raising the flow rate of the oil through the heat exchanger.

While turbulence promoters and extended surfaces are sometimes provided in heat exchangers (such as finning, dimpling, etc.) to reduce air-side insulating boundary effects and to increase heat transfer area, the bores of the air tubes of the proposed unit are clean and smooth. The air mass velocity is sufficiently high so that turbulence within the air tubes is assured. The analysis indicates that any discontinuities in the flow stream far outweigh any advantages by detrimentally increasing pressure drop through the tubes.

Some leeway has been taken on the oil side by closely packing the tubes, which provides a relatively high oil-side pressure drop through the shell. Oil-side pressure head is not critical and does not "cost" as much in terms of weight, performance, or design modification as air-side pressure head does.

The oil-side baffling is of the segmented type, as shown in Figure 10, designed to pass the oil across the air tubes in a normal or transverse direction for maximum dwell time and heat transfer. The oil must make seven transverse passes (six baffles), which takes more time than to make one parallel pass for the unbaffled configuration.

TABLE VIII. PERFORMANCE PARAMETERS - CH-54A ROTOR SHAFT COOLING SYSTEM

Parameter	Value
Oil Temperature Into Heat Exchanger (^{o}F)	280
Oil Temperature Into Gearbox (^{o}F)	246
Heat Rejected Through Housings (Btu/min)	1,195
Air Mass Flow Rate (lb/hr)	6,900
Oil Mass Flow Rate (gpm)	30
Heat Exchanger Effectiveness	0.8
Air-Side Heat Transfer Area (ft^2)	61.5
Oil-Side Heat Transfer Area (ft^2)	67.0
Cross-Sectional Air-Side Area (ft^2)	.121
Air Mass Velocity (lb/hr ft^2)	57,200
Air-Side Reynolds Number	10,500
Air-Side Film Heat Transfer Coefficient (Btu/hr ft^2 ^{o}F)	54.2
Oil Mass Velocity (lb/hr ft^2)	1,060,000
Oil-Side Reynolds Number	1,120
Oil-Side Film Heat Transfer Coefficient (Btu/hr ft^2 ^{o}F)	460
Overall Heat Transfer Coefficient (Btu/hr ft^2 ^{o}F)	48.5
Calculated Heat Exchanger Core Length (ft)	1.22
Total Air-Side Suction Head (in. of H_2O)	61.9
Blower Shaft Horsepower (HP)	33.4
Oil-Side Pressure Drop (psi)	32

TABLE IX. WEIGHT SUMMARY - CH-54A ROTOR SHAFT COOLING SYSTEM	
Cooling System Subcomponent	Weight (lb)
Heat Exchanger, Stainless Steel	31.5
Heat Exchanger Support Structure	9.1
Air-Side Ducting	10.0
Oil-Side Plumbing and Accessories	2.3
Blower	22.7
Blower Drive and Shafting	18.8
Total System Dry Weight	94.4

The heat exchanger is a counterflow device which is the most effective type of heat exchanger obtainable in any core configuration; that is, the cool air enters at the top of the exchanger and proceeds downward through the air tubes, and the hot oil enters at the bottom of the exchanger and progresses upward through the shell. Even though the oil passes transversely across the tube bank, its net progression is upward through the shell, or counter to the air in flow direction. Sometimes referred to as cross-counterflow exchangers, the baffled tube bundle approaches pure counterflow as the number of baffles increases. For this configuration, six baffles on 2-inch spacing definitely is considered to be dense baffling, and the flow situation is considered to be counterflow.

The counterflow exchanger is the most effective configuration because a relatively constant temperature differential is maintained between the hot stream and the cool stream throughout the length of the exchanger, as shown in Figure 13. Cool air entering is next to cooler oil exiting from the exchanger, and warm air exiting is next to hot oil entering the exchanger; this relation is maintained through every point along the exchanger length.

A relatively severe pressure loss is sustained at the inlet to the heat exchanger, where the sudden contraction effect of transition from a 6.8-inch-diameter duct to a flat plate containing 1680 .115-inch-inside-

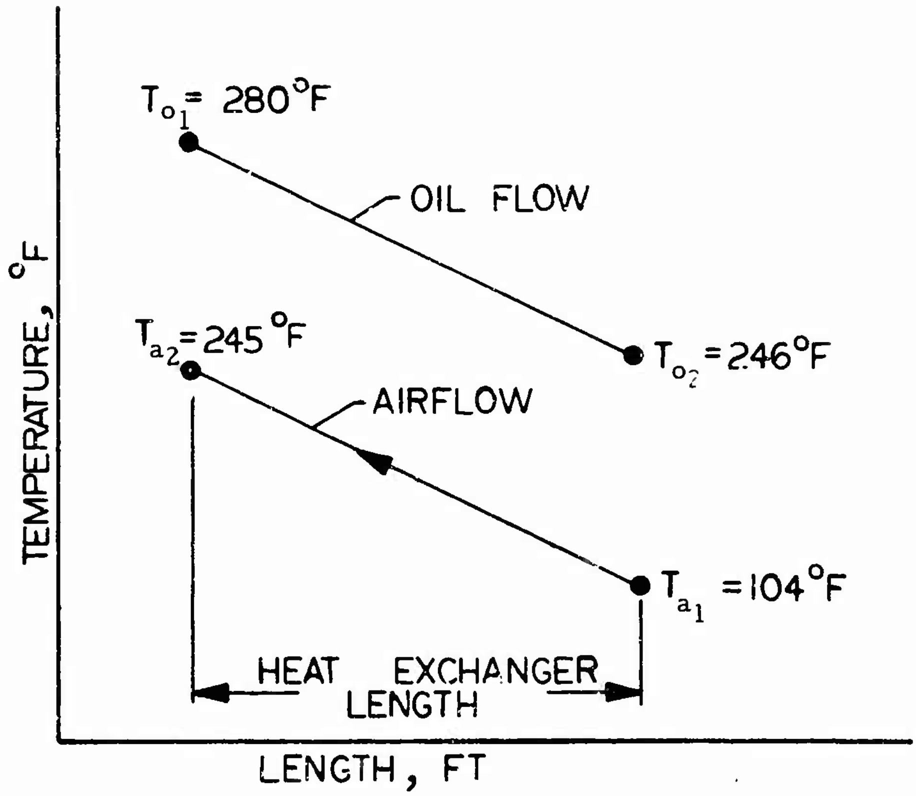


Figure 13. Heat Flow Diagram for Counterflow.

diameter air tubes is imposed on the airstream. This loss is calculated and is minimized by maximizing the exchanger air-side cross-sectional frontal area. Efforts to conceive an intricate inlet streamlining device for every air tube were discouraged by vendor consultation. The cost of fabrication of this device apparently outweighs the aerodynamic benefit.

OIL SUMP COOLING SYSTEM

System Description

In this system, a compact finned-plate heat exchanger is mounted integrally with the oil sump of the CH-54A main gearbox. Oil circuit plumbing from the lubrication pump and to the distribution system is conducted to the exchanger inlet and outlet manifolds. A shaft-driven axial blower delivers ambient air from one side of the transmission/cargo hoist bay area, through the heat exchanger, and discharges it to the other side of the bay area. The system is shown in Figure 14.

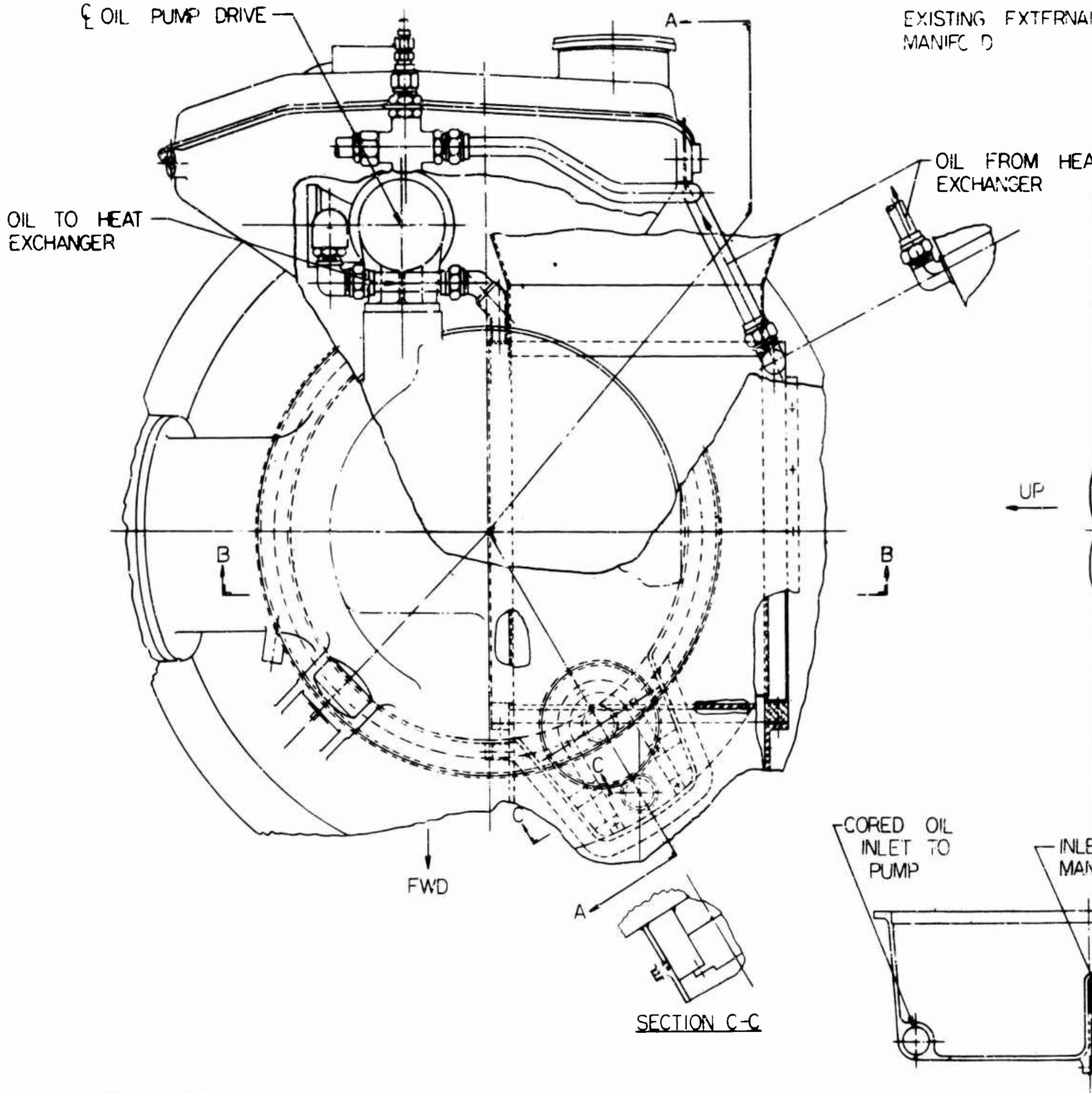
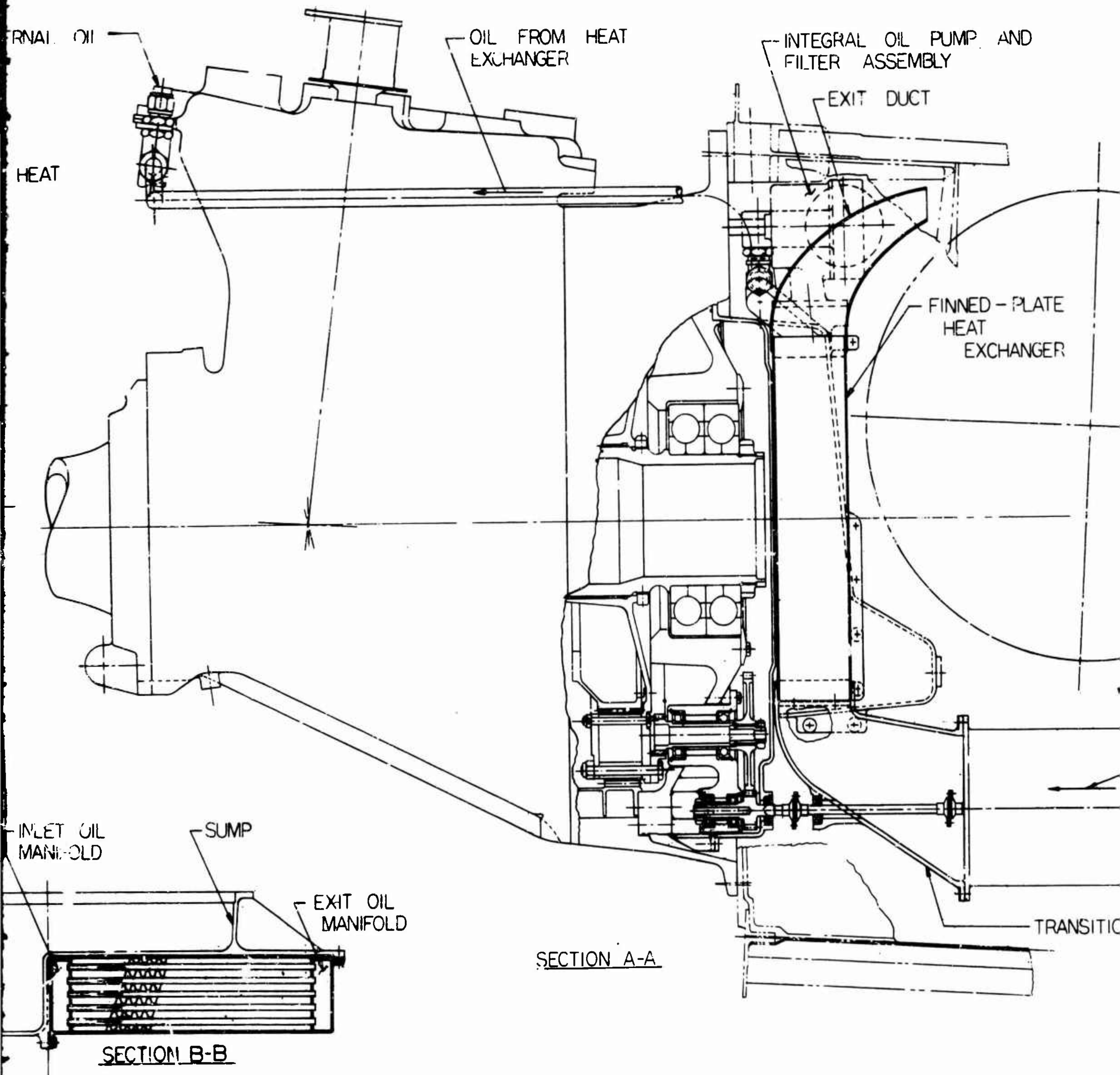
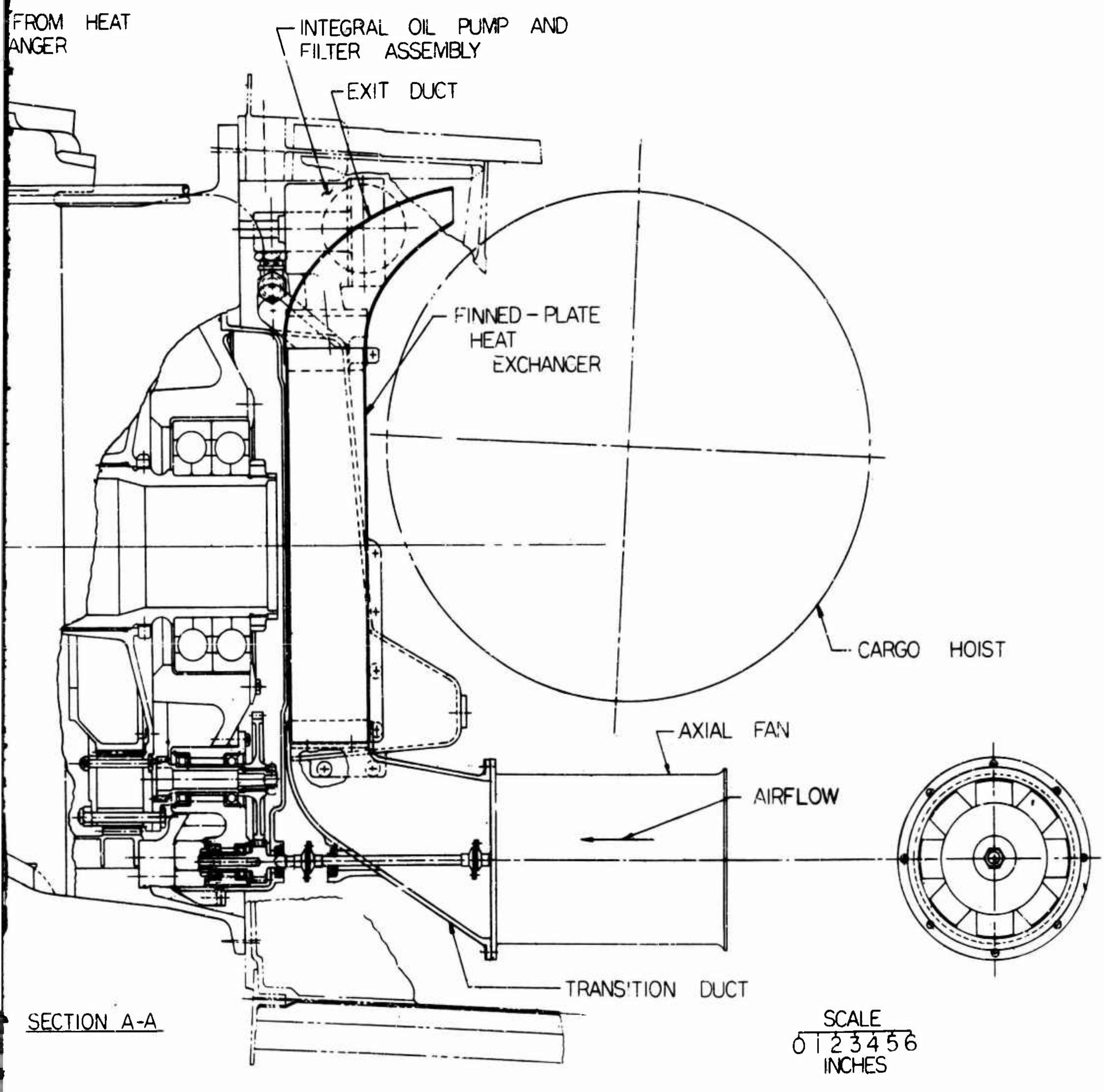


Figure 14. CH-54A Oil Sump Cooling System.


RNAL OIl
HEAT
OIL FROM HEAT EXCHANGER
INTEGRAL OIL PUMP AND FILTER ASSEMBLY
EXIT DUCT
FINNED-PLATE HEAT EXCHANGER
INLET OIL MANIFOLD
SUMP
EXIT OIL MANIFOLD
SECTION A-A
TRANSITIC
SECTION B-B


FROM HEAT
ANGER
INTEGRAL OIL PUMP AND
FILTER ASSEMBLY
EXIT DUCT
FINNED - PLATE
HEAT
EXCHANCER
CARGO HOIST
AXIAL FAN
AIRFLOW
TRANSITION DUCT
SCALE
0 1 2 3 4 5 6
INCHES

SECTION A-A

Heat Exchanger

Both a finned tube and a finned-plate core configuration were analyzed for adaptability to a rectangular heat exchanger envelope mounted integrally as part of the oil sump. The air flows through the core longitudinally, giving rise to a relatively high air-side pressure drop. While the finned-tube core is slightly lighter in weight, the finned-plate design has a lower pressure drop and a lower calculated blower power requirement. For these reasons, the finned-plate configuration was chosen for the preliminary design.

The core dimensions, excluding oil-side headers, are frontal area of 59.7 square inches (14 inches by 4.25 inches) and length of 15 inches. The exchanger is fabricated entirely from aluminum alloy plates and fins dip-brazed together. The core consists of alternate air and oil passages separated by finned plates .020 inch thick. There are eight air passages .405 inch wide and seven oil-side passages .085 inch wide. The outer shell plates are .060 inch thick. The heat exchanger operates on the crossflow principle since the oil flows transversely to the airflow. Thin aluminum zig-zag finning is employed on both the air side and the oil side to increase heat exchanger efficiency.

The heat exchanger is bolted to the oil sump and may be separated from the sump if necessary to provide better maintainability.

Axial Blower

An axial blower provides cooling air to the exchanger. Direct shaft drive is utilized to the blower from the gearbox as shown in Figure 14. The blower is a single-stage unit requiring one set of stator and rotor vanes, fabricated entirely from aluminum with the exception of the sealed bearings and shaft. The blower operates in the "pusher" mode because at the relatively low pressure head through the exchanger, the work done on the air by the blower does not result in a significantly large increase in air-stream temperature from adiabatic heat of compression. The analysis also indicates a lower blower power requirement for the "pusher" mode than for the draw-through mode.

Blower Drive

A compact step-up gear train employing straight spur gears driven by the main gearbox second-stage planetary carrier drives the axial blower by means of a drive shaft. Flexible disc couplings at either end of the shaft accommodate parallel and angular misalignment. The blower speed is 9000 rpm.

Ducting and Control Valves

The inlet duct to the heat exchanger is a smooth transition from the round discharge section of the blower to the rectangular inlet section to the heat exchanger. The duct changes the direction of the airflow to facilitate mounting of the blower and drive shafting to the blower. The exit duct provides a streamlined discharge for the air back into the atmosphere as well as directs the flow downward. The cross-sectional area of the exit duct at the discharge is greater than the cross-sectional area at the heat exchanger outlet, providing a diffuser effect to reduce turbulence and exit losses. A thermostatic bypass valve is incorporated into the oil outlet header to permit sufficient oil flow during cold weather operation to eliminate any congealing tendency of the oil.

Lubrication System Operation

The analysis indicates, as it did for the main shaft heat exchanger, that both the air-side and the oil-side performance is improved when the mass rate of flow is increased from 17 gpm to 30 gpm. The pressure regulator and bypass valve are placed after the heat exchanger flow-wise instead of before the heat exchanger, so that all the pump flow passes through the heat exchanger. Then 17 gpm flows to the distribution system and 13 gpm is bypassed directly back to the sump.

Performance Analyses

Design Analysis

The calculated performance of the oil sump cooling system is presented in Table X. The analysis for this system is similar to that made for the main shaft cooler in Appendix I, and therefore the detailed calculations are not repeated.

Weight Analysis

The calculated weights for the component parts of the oil sump cooling system are summarized in Table XI.

VAPOR CYCLE REFRIGERATION SYSTEM

System Description

The vapor cycle refrigeration system is a closed-loop heat transport system operating between the hot oil in the gearbox and the ambient atmosphere. Several major subcomponents are located integrally as part of the transmission system or in the transmission/cargo hoist bay area and are

TABLE X. PERFORMANCE PARAMETERS - CH-54A OIL SUMP COOLING SYSTEM

Parameter	Value
Oil Temperature Into Heat Exchanger (^oF)	280
Oil Temperature Into Gearbox (^oF)	246
Ambient Air Temperature, Maximum (^oF)	104
Adiabatic Heat of Compression Across Blower (^oF)	21
Actual Air Temperature Into Heat Exchanger (^oF)	125
Air Temperature out of Heat Exchanger (^oF)	202.5
Heat Exchanger Heat Load (Btu/min)	3,905
Air Mass Flow Rate (lb/hr)	12,540
Oil Mass Flow Rate (gpm)	30
Heat Exchanger Effectiveness	.5
Air-Side Heat Transfer Area (ft^2)	89.6
Oil-Side Heat Transfer Area (ft^2)	44.2
Cross-Sectional Air-Side Area (ft^2)	0.295
Air Mass Velocity (lb/hr ft^2)	42,600
Air-Side Reynolds Number	14,900
Air-Side Heat Transfer Coefficient (Btu/hr ft^2 ^oF)	39
Air Velocity Through Heat Exchanger (ft/sec)	185
Oil Mass Velocity (lb/hr ft^2)	176,000
Oil-Side Reynolds Number	560
Oil-Side Heat Transfer Coefficient (Btu/hr ft^2 ^oF)	200
Overall Heat Transfer Coefficient (Btu/hr ft^2 ^oF)	32.5
Calculated Heat Exchanger Length (ft)	1.4
Total Air-Side Pressure Drop (in. of H_2O)	30
Blower Air Horsepower (HP)	14.1
Blower Shaft Horsepower (HP)	17.5
Oil-Side Pressure Drop (psi)	13

TABLE XI. WEIGHT SUMMARY - CH-54A OIL SUMP COOLING SYSTEM

Cooling System Subcomponent	Weight (lb)
Heat Exchanger, Aluminum	23.8
Heat Exchanger Support Structure	.5
Air-Side Ducting	6.5
Oil-Side Plumbing and Accessories	2.9
Blower	20.0
Blower Drive and Shafting	15.0
Total System Dry Weight	68.7

plumbed together to form the refrigerant circuit. Refrigerant is circulated in vapor form by the compressor to the condenser, where system heat is rejected to ambient air supplied by the condenser blower. The vapor is changed to liquid in the condenser, where it is pumped to the receiver and to the throttle. In the throttle, the refrigerant temperature is reduced by expansion to the evaporator temperature level. The liquid is then circulated to the evaporator, where gearbox oil heat is absorbed by changing the state of the refrigerant from liquid to vapor. The vapor is pumped back to the compressor to complete the cycle. The refrigerant charge used in the system is Refrigerant 12. The system is shown in Figure 15.

Compressor

The compressor selected for the system is a high-speed helical screw rotor design developed specifically for airborne use. The compressor envelope is 7 inches in diameter and 8 inches long. The compressor is mounted on the accessory cover of the main gearbox and is direct shaft driven at 12,500 rpm. An additional accessory pad for the compressor will have to be provided on the rear cover.

Compressor power requirement is 25 HP. The compressor is fabricated from stainless steel and aluminum, which are compatible with the refrigerant. Lubrication for the compressor is provided by a sealed, self-contained oil charge. No maintenance is required.

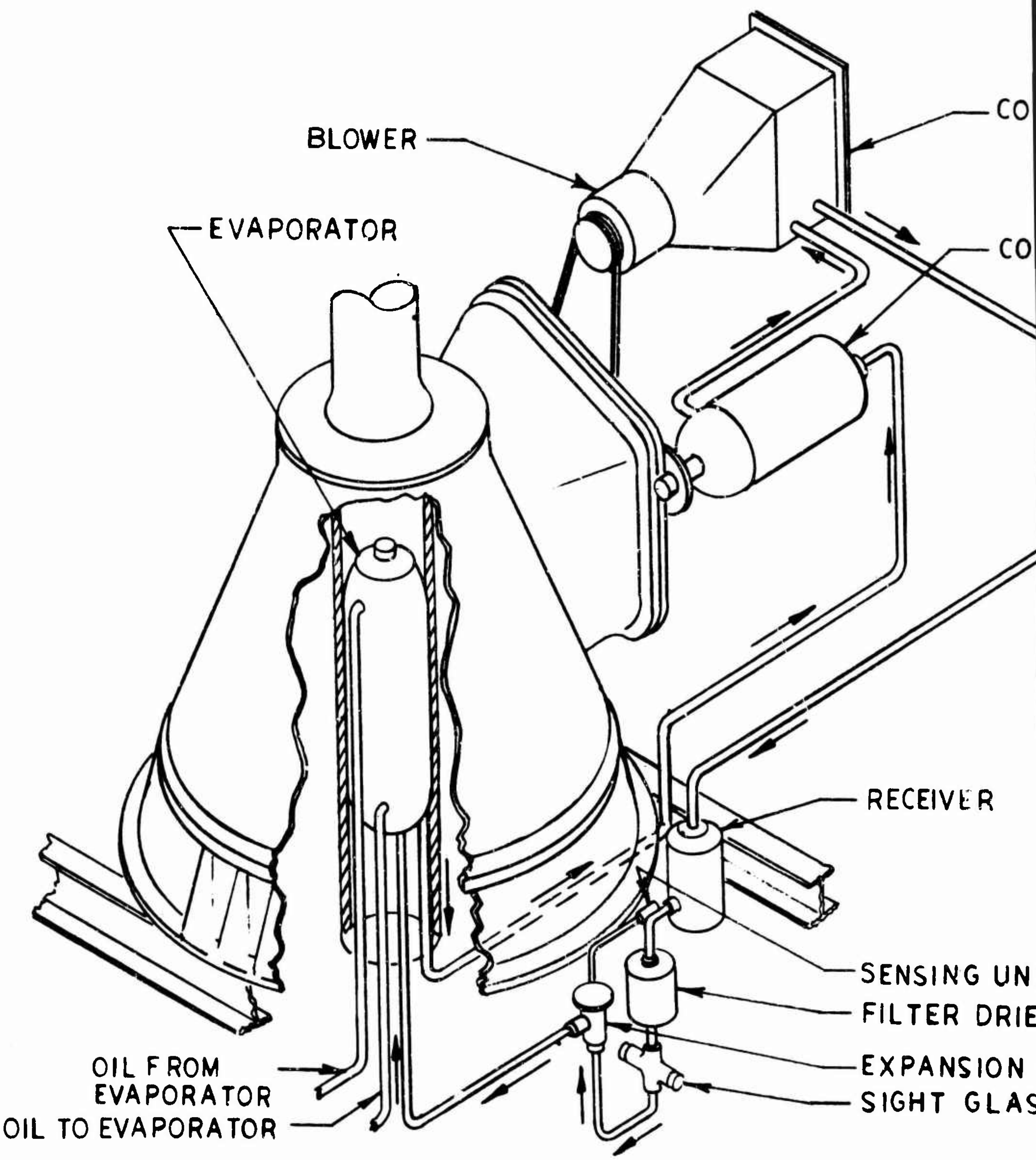
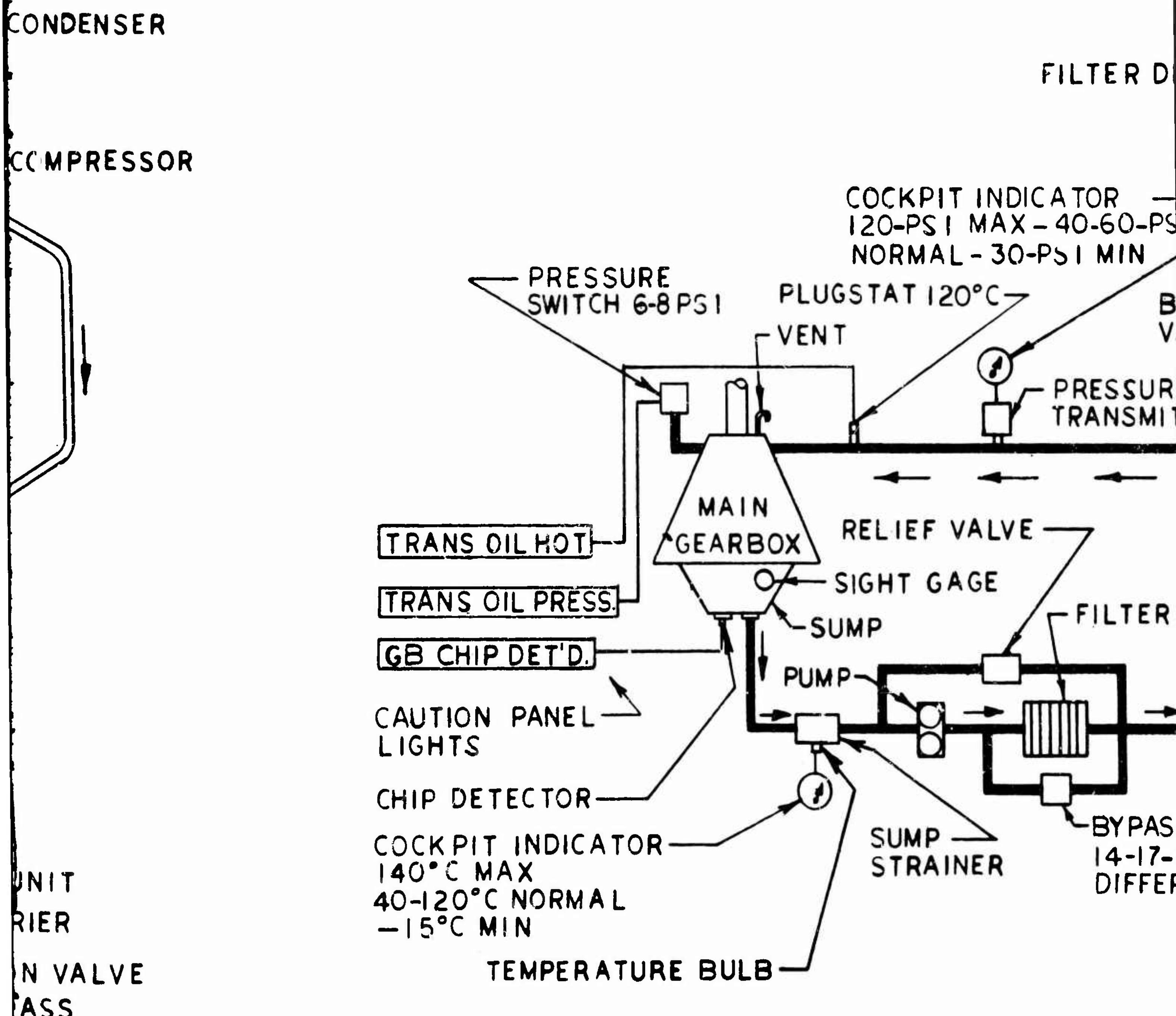
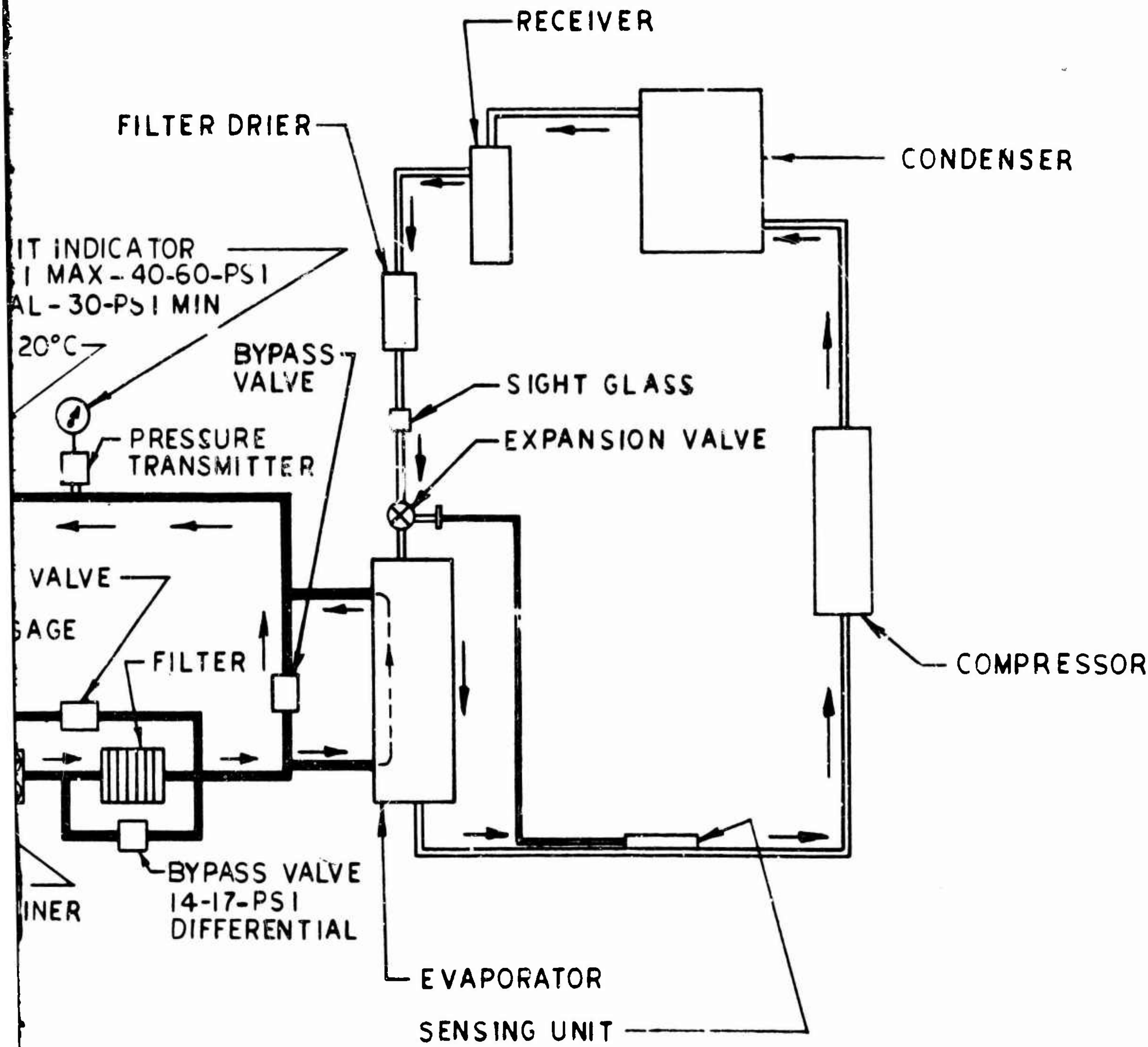


Figure 15. Vapor Cycle Cooling System Schematic.

A



CONDENSER
CCMPRESSOR
FILTER D
COCKPIT INDICATOR
120-PSI MAX - 40-60-PS
NORMAL - 30-PSI MIN
PRESSURE
SWITCH 6-8 PSI
PLUGSTAT 120°C
VENT
PRESSUR
TRANSMI
MAIN
GEARBOX
TRANS OIL HOT
TRANS OIL PRESS.
GB CHIP DET'D.
RELIEF VALVE
SIGHT GAGE
SUMP
FILTER
PUMP
CAUTION PANEL
LIGHTS
CHIP DETECTOR
COCKPIT INDICATOR
140°C MAX
40-120°C NORMAL
—15°C MIN
SUMP
STRAINER
BYPASS
14-17-
DIFFER
TEMPERATURE BULB
UNIT
RIER
N VALVE
ASS


RECEIVER
FILTER DRIER
CONDENSER
IT INDICATOR
I MAX - 40-60-PSI
AL - 30-PSI MIN
20°C
BYPASS VALVE
SIGHT GLASS
PRESSURE TRANSMITTER
EXPANSION VALVE
VALVE
AGE
FILTER
COMPRESSOR
BYPASS VALVE 14-17-PSI DIFFERENTIAL
INER
EVAPORATOR
SENSING UNIT

Condenser

The condenser is an aluminum rectangular plate-fin high-pressure refrigerant-to-air heat exchanger mounted externally to the rear portion of the main gearbox and supported similarly to the current CH-54A oil cooler. The condenser envelope is 10 inches by 17 inches by 8 inches deep. Appropriate transition ducting from the heat exchanger to the inlet of the blower is provided.

Condenser Blower

The ambient air is drawn through the condenser by the condenser blower. The draw-through system, instead of the "pusher" system, is employed in the refrigeration system condenser blower for the same reasoning that the main shaft cooling system blower operates on the draw-through principle, as discussed in the preliminary design section of that system. The blower is a multiple-stage axial blower that is belt or shaft driven from the accessory section of the main gearbox. Condenser blower power requirement is 30 HP. An additional power takeoff pad on the gearbox rear cover is required to accommodate the blower drive.

Receiver

The refrigerant receiver or reservoir is a significant system accessory required to ensure adequate refrigerant supply. The receiver is located in the transmission/cargo hoist bay area and is plumbed between the condenser and the throttle. Envelope dimensions of the receiver are 8 inches in diameter by 24 inches in length.

Throttle (Expansion Valve)

The throttle selected for this system is a thermostatic control element which meters refrigerant into the evaporator as required, depending upon the degree of refrigerant superheat detected by the sensing bulb located in the evaporator outlet manifold. Envelope dimensions of the throttle are 5 inches by 4 inches by 3 inches. The throttle is mounted in the transmission/cargo hoist bay area of the CH-54A.

Evaporator

The evaporator is a shell and tube-type oil-to-refrigerant heat exchanger. For maximum security from battle damage, the evaporator is mounted within the main rotor shaft, similarly to the heat exchanger of the main shaft cooling system. Refrigerant flows through the tubes in a two-pass circuit, necessitating only one refrigerant header on one end. The second or downflow pass provides greater flow area than the upward pass to

accommodate increasing vapor flow velocity and to prevent vapor lock. Oil flows in a well-baffled (6 baffles) shell-side circuit. Inlet and outlet refrigerant ports are located in the lower spherical end dome manifold. Inlet and outlet oil ports are located at the top and at the bottom of the outside diameter of the shell. Clearance between the shell and the main shaft is sufficient to accommodate the oil lines and fittings. The overall evaporator is 5 inches in diameter by 17 inches in length. The evaporator is of all-aluminum construction with dip-brazed tube-to-end plate connections.

Other Subcomponents

All the accessories normally required for a vapor cycle refrigeration system are included, such as filter drier, liquid level monitor, service valves, and all system plumbing.

Refrigerant

The heat transport fluid contained in the closed system is Refrigerant 12. Several refrigerants were analyzed, and reasons for selecting Refrigerant 12 are contained in Appendix II.

Performance Analysis

Design Analysis

The analysis of the vapor cycle refrigeration system is presented in Appendix II. The analysis was performed for several condensing temperatures and several saturated suction temperatures. The analysis in the appendix is conducted for the final solution used in the preliminary design.

Weight Analysis

The calculated weights of each subcomponent of the refrigeration system are summarized in Table XII. System dry weight and refrigerant charge weight are included.

TABLE XII. WEIGHT SUMMARY - VAPOR CYCLE REFRIGERATION SYSTEM

Subcomponent	Weight (lb)
Compressor	15.0
Compressor Drive	5.3
Condenser and Support Structure	33.0
Condenser Transition Ducting	3.0
Condenser Blower	15.0
Condenser Blower Drive	21.7
Receiver and Support	16.1
Throttle	5.0
Evaporator	12.0
Evaporator Support Structure	8.0
System Accessories	8.0
System Plumbing	8.7
Total System Dry Weight	150.8
Refrigerant Charge	45.0
Total	195.8

RELIABILITY, VULNERABILITY, AND MAINTAINABILITY ANALYSIS

INTRODUCTION

A comparative reliability, vulnerability, and maintainability study was made of the current production CH-54A main gearbox cooling system and the integral cooling systems of this report. The study was based on the system descriptions and layout drawings of the Preliminary Design section. Estimates of mean time between failures, maintenance man-hours, and percentage of cooling system small-arms "hits" have been made. A failure mode and effect analysis was also performed.

ANALYSIS

Reliability

The components of the proposed cooling systems have predicted failure rates based on similar component reliability data from other model experience and vendor consultation, as shown in Table XIII.

Estimated MTBFs are calculated by taking the reciprocal of the summation of the individual component failure rates and are presented in Table XIII for comparison.

Static components such as supporting structure, heat exchangers, and most hardware are relatively immune to environment and operating conditions. Failures of these components are usually random, and MTBF values are high. Control devices such as thermostatic regulators, expansion valves, and pressure switches are dynamic and susceptible to cyclic fatigue, since they contain springs, diaphragms, and bimetallic elements. Failure of these elements is dependent upon the number of cycles, which is a function of operating conditions. Rotating components such as blowers, blower drives, and compressors operate at high speeds and have the lowest life expectancy, mainly because of the rolling-element bearings used. Reliability of these components can be increased if rotative speed can be decreased. Speeds in all the designs proposed are minimized as much as practicable.

Failure Mode and Effect

The results of a failure mode and effect analysis are summarized in Table XIV. Only the most probable failure modes, as indicated by the reliability analysis, are considered. Design features that minimize failure within these modes are also included in Table XIV.

Vulnerability

A comparative vulnerability analysis was conducted for the proposed integral cooling systems and the current production CH-54A cooling configuration. A discussion of the design features that affect vulnerability/invulnerability of each of the studied systems follows. Table XV is a summary of the degree of both the relative oil circuit and the cooling circuit and small-arms protection available. The estimated percentage of oil circuit and cooling circuit hits for each system is also included. Armor plate analysis is not included in this study. All vulnerability calculations are for unarmored configurations.

In judging relative vulnerability, a hit sustained by the cooling circuit (e.g., blower or blower drive) is weighted the same as a hit sustained by the oil circuit. While the transmission may function long enough to effect emergency evasive action if cooling capability is lost but oil lubrication capability is not lost, the system is considered to be vulnerable. Protecting the oil circuit alone is not a goal of this study.

Current Production CH-54A Configuration

While as of this writing no CH-54A aircraft have been lost as a result of sustaining a hit in the main transmission cooling system, the current production cooling configuration is somewhat vulnerable to small-arms fire.

The current production system consists basically of a plate-fin oil-to-air heat exchanger mounted on the rear cover section of the main gearbox with struts and brackets. A belt-driven blower is mounted to the heat exchanger by an inlet transition duct. Blower drive is provided by the tail rotor drive shaft through pulleys and V-belting.

To augment vulnerability data on the CH-54A because of its relative newness to the field, combat damage information from the HH-3E and CH-3E helicopter was used Based on the projected vulnerable areas of the cooling system components, the percentage of the total number of aircraft hits sustained by any portion of the cooling system is predicted.

The heat exchanger and blower installation are protected only for small-arms trajectories from below by the transmission deck and airframe and from forward by the main gearbox itself. Out of 10,000 hits to the aircraft, it is calculated that 11 will strikes the transmission cooling system.

Rotor Shaft Cooling System

The integral heat exchanger located within the main rotor shaft, including inlet and outlet oil plumbing, is virtually invulnerable to small-arms fire,

since .30 and .50 caliber projectiles will not penetrate the wall of the main shaft. The oil circuit has a perfect predicted invulnerability. There is a slight probability that the blower could sustain a hit for trajectories straight up from below. However, it is calculated that only 1.5 hits out of 10,000 received by the aircraft will strike the blower.

Maintainability

Using accumulated data for cooling system removal and installation man-hours required on the CH-54A as well as the CH-53A series helicopters, maintenance man-hour estimates were made and are presented in Tables XVI and XVII. The total maintenance man-hour estimates are summarized in Table XVIII.

The rotor shaft cooling system is by far the most integral and most invulnerable system investigated under the scope of this study.

Oil Sump Cooling System

The finned-plate heat exchanger mounted to the oil sump is protected from most trajectories by the cargo hoist assembly and the hoist bay bulkheads, and it is therefore not completely invulnerable. It is calculated that 4.5 hits out of 10,000 will strike the heat exchanger. A small amount of strategically placed armor plate will reduce this vulnerability to zero. The blower is located in the same manner as the blower of the main shaft cooling system and therefore has the same calculated vulnerability. Predicted system vulnerability reveals that 6.0 hits out of 10,000 total aircraft hits will strike the oil sump cooling system.

Vapor Cycle Refrigeration System

The oil circuit portion of the vapor cycle refrigeration system, made up of the evaporator (oil-to-refrigerant heat exchanger) and associated oil inlet and outlet plumbing, like the main shaft cooling system, is completely buried within, and protected by, the main rotor shaft.

The condenser and condenser blower assembly, as well as the compressor and associated system plumbing, cannot be located integrally or within the confines of the hoist bay because of its large size and direct drive requirements. These components are therefore located on the rear cover section of the main gearbox, and they have vulnerabilities greater than the current production CH-54A components, due to their large projected areas. Out of 10,000 hits received by the aircraft, 17 hits are calculated to strike these components. Vulnerability of the refrigeration system is further increased by the receiver, throttle, and associated accessories required to complete the installation.

TABLE XIII. COMPARATIVE SUBCOMPONENT FAILURE RATES AND TOTAL SYSTEM ESTIMATED MEAN TIME BETWEEN FAILURES

Parameter	Rotor Shaft Cooling System		Oil Sump Cooling System		Vapor Cycle Refrigeration System		Current Production CH-54A Configuration	
Subcomponent Failure Rate*	Blower	5.00	Blower	2.50	Evaporator	.15	Blower	1.50
	Heat Exchanger	1.00	Heat Exchanger	.50	Condenser	.15	Radiator	.10
	Blower Drive	2.50	Blower Drive	2.50	Condenser Blower	15.00	Blower Drive	1.50
	Top Support	1.00	Exchanger Support	.05	Receiver	.10	Support Structure	.05
	Exchanger Support	.10	Blower Support	.50	Throttle	.68	Misc Hardware	.05
	Blower Support	1.25	Misc Hardware	.05	Compressor	9.80		
	Internal Plumbing	.10			Misc Hardware	.05		
	Misc Hardware	.05						
Total Failure Rate		11.00		6.10		25.91		3.20
MTBF (hr)		9,100		16,400		3,850		31,500

*Per 10,000 hours total population time

TABLE XIV. FAILURE MODE AND EFFECT ANALYSIS

System	Component	Failure Mode	Effect	Minimizing Design Feature
Rotor Shaft Cooling System	Heat Exchanger	Oil-side leakage due to cracked tube or brazing deficiency	None unless allowed to progress to point of depleting gearbox oil supply	The shell and tubes are both fabricated from stainless steel for strength, compatibility, and resistance to vibration. Dip-brazing assembly technique minimizes possibility of leakage.
	Heat Exchanger Upper Support Bearing	Seal or prepacked lubricant failure	None unless allowed to progress to ball retainer failure and contamination of gearbox sump	Bearing operates at only 185 rpm and has a very high, calculated B-10 cyclic life.
	Heat Exchanger Inlet and Outlet Plumbing	Cutting of O-ring during installation at sump-to-heat exchanger connection	Oil leakage directly back into sump - no loss of oil	O-ring groove and lead-in chamfers are per standards. From experience, no problems are anticipated.
	Oil Sump and Transition Duct	Cracking	Possible oil leakage; structurally, none unless allowed to progress to ultimate fracture	Casting and weldment technique are based on investigators' experience, incorporating ribs, fillets, etc.
	Blower	Spalled bearing	Loss of cooling capability if progresses to ball or retainer fracture	Sealed, prepacked bearings are employed throughout. Calculated bearing B-10 life is very high.
	Blower	Flexible disc coupling lamination cracking	None unless all laminations progress to ultimate fracture	The flexible disc coupling is redundant. Alternate disc grain directions are at right angles to each other.

TABLE XIV - Continued

System	Component	Failure Mode	Effect	Minimizing Design Feature
Oil Sump Cooling System	Heat Exchanger	Oil-side leakage	None unless allowed to progress to point of depleting gearbox oil supply	The plates and fins are fabricated from aluminum, dip-brazed together for integrity and compatibility. Shell support to oil sump is redundant. AN and MS fittings and O-rings are employed throughout.
	Heat Exchanger Inlet and Outlet Plumbing	Oil-side leakage	None unless materially affects gearbox oil quantity	
	Blower Support and Transition Duct	Cracking	None unless allowed to progress to ultimate fracture and loss of blower	Structure is conservatively designed to resist vibration.
	Blower	Spalled bearing	None unless allowed to progress to ball or retainer fracture and loss of cooling capability	Sealed, prepacked bearings are employed throughout. Calculated bearing B-10 life is very high.
	Blower Drive	Flexible disc coupling lamination cracking	None unless all laminations progress to ultimate fracture	The flexible disc coupling is redundant. Alternate disc grain directions are at right angles.
Vapor Cycle Refrigeration System	Evaporator	Oil or refrigerant leakage due to cracked tube brazing deficiency	Oil side: None unless allowed to progress until gearbox oil level materially affected Refrigerant side: None unless accompanied by loss of high pressure in system	Shell and tube evaporator is fabricated entirely from stainless steel for strength, compatibility, and resistance to vibration. Dip-brazing technique minimizes possibility of leakage.
	Condenser	Refrigerant leakage due to brazing deficiency or core cracking	None unless accompanied by loss of pressure in system	All-aluminum condenser core is designed for high-pressure operation. Dip-brazing is employed for integrity and minimized leakage.

TABLE XIV - Continued

System	Component	Failure Mode	Effect	Minimizing Design Feature
Vapor Cycle Refrigeration System (continued)	Condenser Blower	Spalled bearing	Loss of cooling capability if progresses to ball or retainer fracture	Sealed, prepacked bearings are employed throughout.
	Condenser Blower Drive	Flexible disc coupling lamination cracking	None unless all laminations progress to ultimate fracture	Inherently redundant coupling design.
	Compressor	Spalling or lubrication failure of compressor bearing	Loss of cooling capability if progresses to point of ball or retainer fracture	All compressor bearings are sealed and provided with their own lubrication charge.
	Compressor Drive	Bearing spalling in gearbox accessory drive	Gearbox chip detector activated, requiring precautionary landing and mission abortion	Gearbox accessory drive incorporates bearings and gearing of very high calculated life.
	Expansion Valve	Cyclic failure of spring or diaphragm	Loss of cooling capability or at least thermostatic control	Calculated and qualified cyclic performance is far in excess of anticipated cyclic requirements.
	Receiver	Refrigerant leakage of seams or fittings	None unless accompanied by loss of pressure of refrigerant	All-welded construction.
	Plumbing and Closed Loop System Accessories	Refrigerant leakage or loss of operating pressure	Probably none if detected early, unless pressure loss is sustained	Welded or dip-brazed fitting connections are employed where applicable. Reliable high-pressure plumbing and tubing are used throughout.

TABLE XIV - Continued

System	Component	Failure Mode	Effect	Minimizing Design Feature
Current Production CH-54A Cooling System	Heat Exchanger	Clogging of core by grass and foreign objects	Some loss of cooling capability, not usually affecting gearbox operating temperature	-
	Blower	Bearing spalling	None unless allowed to progress until ball or retainer fractures, resulting in loss of cooling capability	Sealed, prelubricated bearings are employed throughout.
	Blower Drive	Belt slippage or breakage; belt tensioner improper operation	None unless both belts fail together; one belt will drive blower; slippage will result in accelerated wear and loss of cooling capability	Redundant two-belt design. Belt tensioner device employs spring-loaded idler pulley to maintain constant, correct belt preload.
	Support Structure	Cracking	None unless progresses to ultimate fracture	Reinforced epoxy fiber glass system is employed for ducting. Cast and tubular heat exchanger support system is redundant.

TABLE XV. COMPARATIVE VULNERABILITY

Evaluation	Rotor Shaft Cooling System	Oil Sump Cooling System	Vapor Cycle Refrigeration System	Current CH-54A Configuration
Oil circuit inherent armor protection	Maximum protection from main rotor shaft	Considerable protection from cargo hoist and cargo hoist bay bulkheads	Maximum protection from main rotor shaft for evaporator	Slight protection from main gearbox housing
Cooling circuit inherent armor protection	Considerable protection from cargo hoist bay bulkheads	Considerable protection from cargo hoist bay bulkheads	Slight protection for condenser, condenser blower, and compressor; considerable protection for throttle and receiver from cargo hoist bay bulkheads	Slight protection from main gearbox housing
Oil circuit percentage of total hits sustained based on projected areas	None	4.5 hits out of 10,000	None to evaporator	8 hits out of 10,000
Cooling circuit percentage of total hits sustained based on projected areas	1.5 hits out of 10,000	1.5 hits out of 10,000	17 hits out of 10,000 to the condenser, condenser blower, or compressor; 1.2 hits out of 10,000 to the throttle or receiver	3 hits out of 10,000
Total percentage of hits to cooling system	1.5 hits out of 10,000	6.0 hits out of 10,000	19.2 hits out of 10,000	11 hits out of 10,000

TABLE XVI. MAINTENANCE ESTIMATES AND REMOVAL TIMES

Function	Rotor Shaft Cooling System			Oil Sump Cooling System			Vapor Cycle Refrigeration System			Current CH-54A Configeration		
	Men	Hours	MH	Men	Hours	MH	Men	Hours	MH	Men	Hours	MH
Remove Hoist Assembly	3	1.5	4.5	3	1.5	4.5	3	1.5	4.5	-	-	-
Disconnect Drive and Remove Blower	2	0.2	0.4	2	0.2	0.4	2	0.5	1	2	0.5	1
Drain Oil, Remove Sump Assembly	2	0.5	1	-	-	-	1	0.5	0.5	-	-	-
Remove Heat Exchanger Assembly	2	0.5	1	2	0.2	0.4	-	-	-	2	0.4	0.8
Remove Evaporator	-	-	-	-	-	-	2	0.5	1	-	-	-
Remove Condenser	-	-	-	-	-	-	2	0.4	0.8	-	-	-
Remove Compressor Assembly	-	-	-	-	-	-	2	1	2	-	-	-
Remove Other Cyclic Equipment	-	-	-	-	-	-	2	1	2	-	-	-
Remove Plumbing, Hardware, etc.	1	0.5	0.5	1	0.2	0.2	1	1	1	1	0.2	0.2
Removal Totals	-	3.2	7.4	-	2.1	5.5	-	6.4	12.8	-	1.1	2

TABLE XVII. MAINTENANCE ESTIMATES AND INSTALLATION TIMES

Function	Rotor Shaft Cooling System			Oil Sump Cooling System			Vapor Cycle Refrigeration System			Current CH-54A Configuration		
	Men	Hours	MH	Men	Hours	MH	Men	Hours	MH	Men	Hours	MH
Install Heat Exchanger Assembly	2	1	2	1	1	1	-	-	-	2	1	2
Install Evaporator	-	-	-	-	-	-	2	1	2	-	-	-
Install Condenser	-	-	-	-	-	-	2	1	2	-	-	-
Install and Service Sump Assembly	2	1	2	-	-	-	1	1	1	-	-	-
Install Blower and Connect Drive	2	1	2	2	1	2	2	1	2	2	1	2
Install Hoist Assembly	3	2	6	3	2	6	3	2	6	-	-	-
Install Compressor Assembly	-	-	-	-	-	-	2	1.5	3	-	-	-
Install Other Cyclic Equipment	-	-	-	-	-	-	2	1.5	3	-	-	-
Install Plumbing, Hardware, etc.	1	1	1	1	0.5	0.5	1	2	2	1	0.5	0.5
Installation Totals	-	6	13	-	4.5	9.5	-	11	21	-	2.5	4.5

Of the three cooling systems investigated in the preliminary design phase, the refrigeration system is the least invulnerable.

TABLE XVIII. TOTAL MAINTENANCE ESTIMATES		
Cooling System	Elapsed Time (hr)	Total Man-Hours
Rotor Shaft Cooling System	9.2	20.4
Oil Sump Cooling System	6.6	15.0
Vapor Cycle Refrigeration System	17.4	34.3
Current CH-54A Configuration	3.6	6.5

ADAPTABILITY TO OTHER MODELS

INTRODUCTION

To determine the suitability of the integral cooling systems of this report to other helicopter gearboxes, an investigation of the CH-54A and UH-1D main transmission and lubrication systems was made. Although the data on the gearboxes of these aircraft available to the investigators were limited, they were sufficient for a conceptual study.

Of the three integral cooling systems investigated for the CH-54A, the main rotor shaft and the oil sump coolers were considered to be the most feasible systems for adaptation to the other aircraft.

CH-47A HELICOPTER

Basic Data

Vehicle Description

The CH-47A, shown in Figures 16 and 17, is a tandem-rotor helicopter powered by two T-55-L-7 shaft turbine engines driving 59-foot-diameter fore-and-aft main rotors.

Drive Train Configuration

Engine torque is transmitted from the engines through engine-mounted angle transmissions, which contain an overrunning one-way sprag clutch to a combining transmission via a short drive shaft. The combining transmission mixes the power delivered from each engine and distributes the combined torque to the forward main transmission via seven sections of drive shafting and to the aft main transmission via two sections of drive shafting. The aft portion of the aft main transmission is an accessory gearbox driven by the aft transmission through an overrunning sprag clutch. An auxiliary power unit drives the accessory gearbox via a hydraulic motor for accessory operation independent of the main rotors. An angle takeoff and a short drive shaft from the aft main transmission provide power to one centrally located blower, which provides cooling air for the cooling systems.

The main transmissions are basically similar and consist of an input bevel stage receiving power from the combining transmission output and driving the main rotor through a two-stage planetary system employing stationary ring gears. A cutaway view of the transmission system schematic and the forward main transmission is shown in Figure 18.

Figure 16. CH-47A Aircraft.

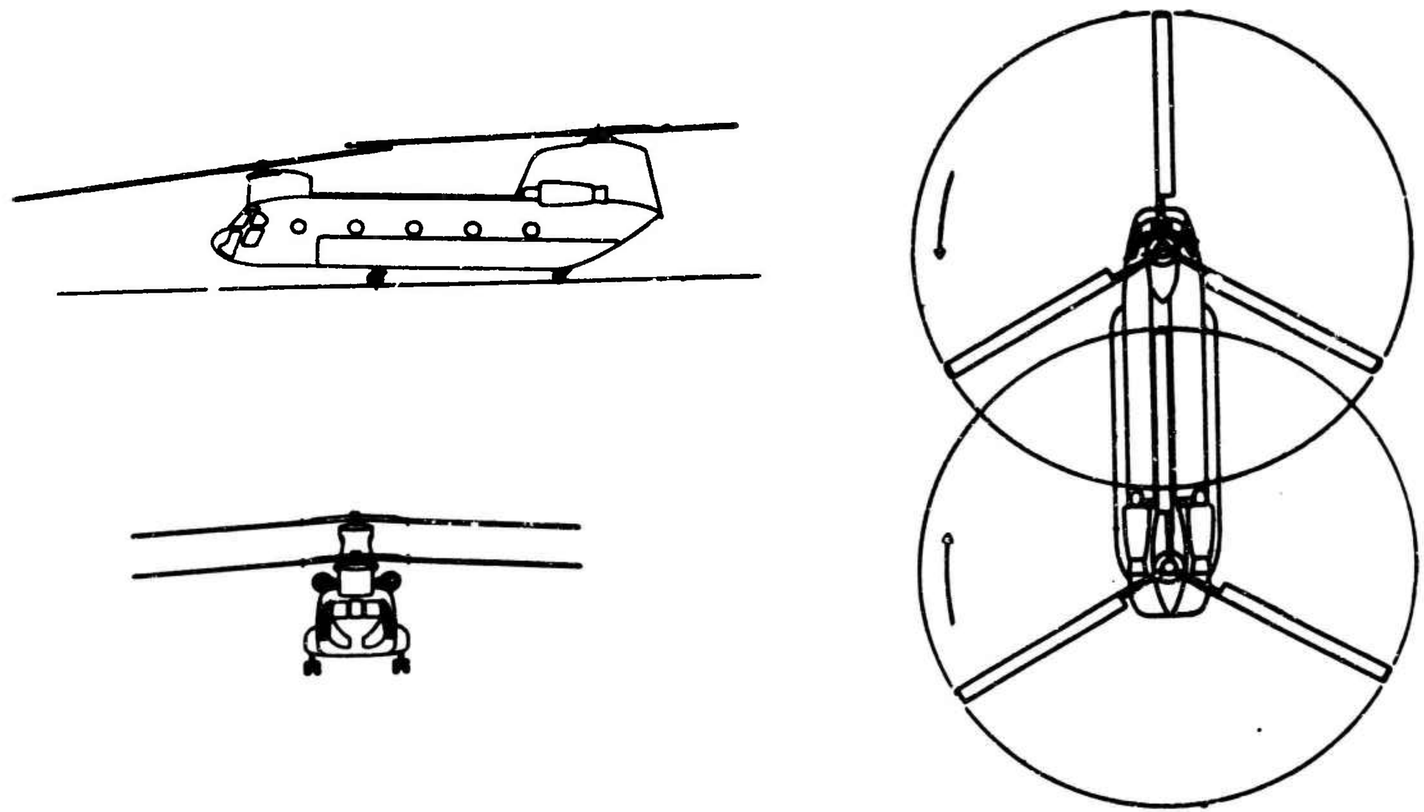


Figure 17. CH-47A Aircraft General Arrangement.

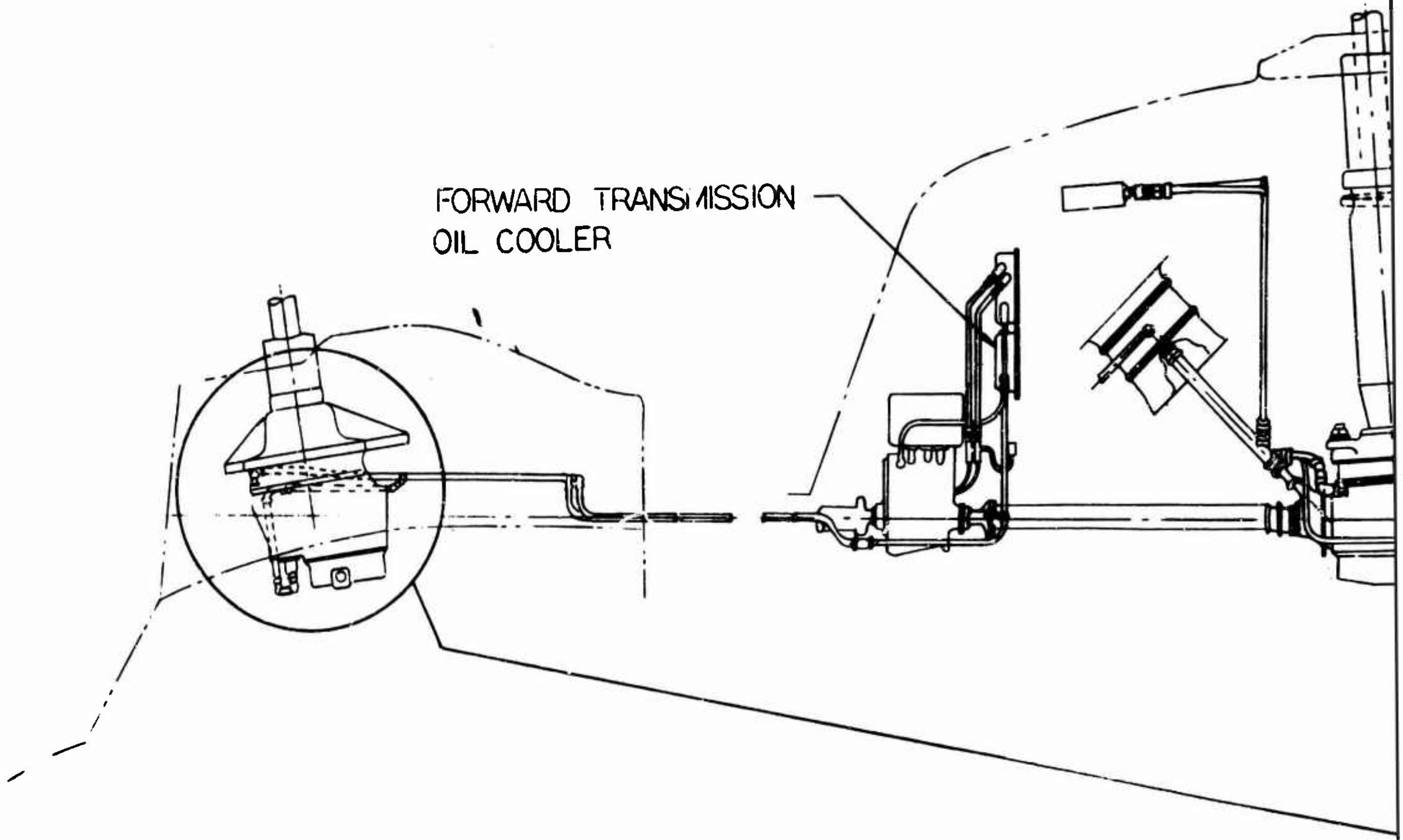
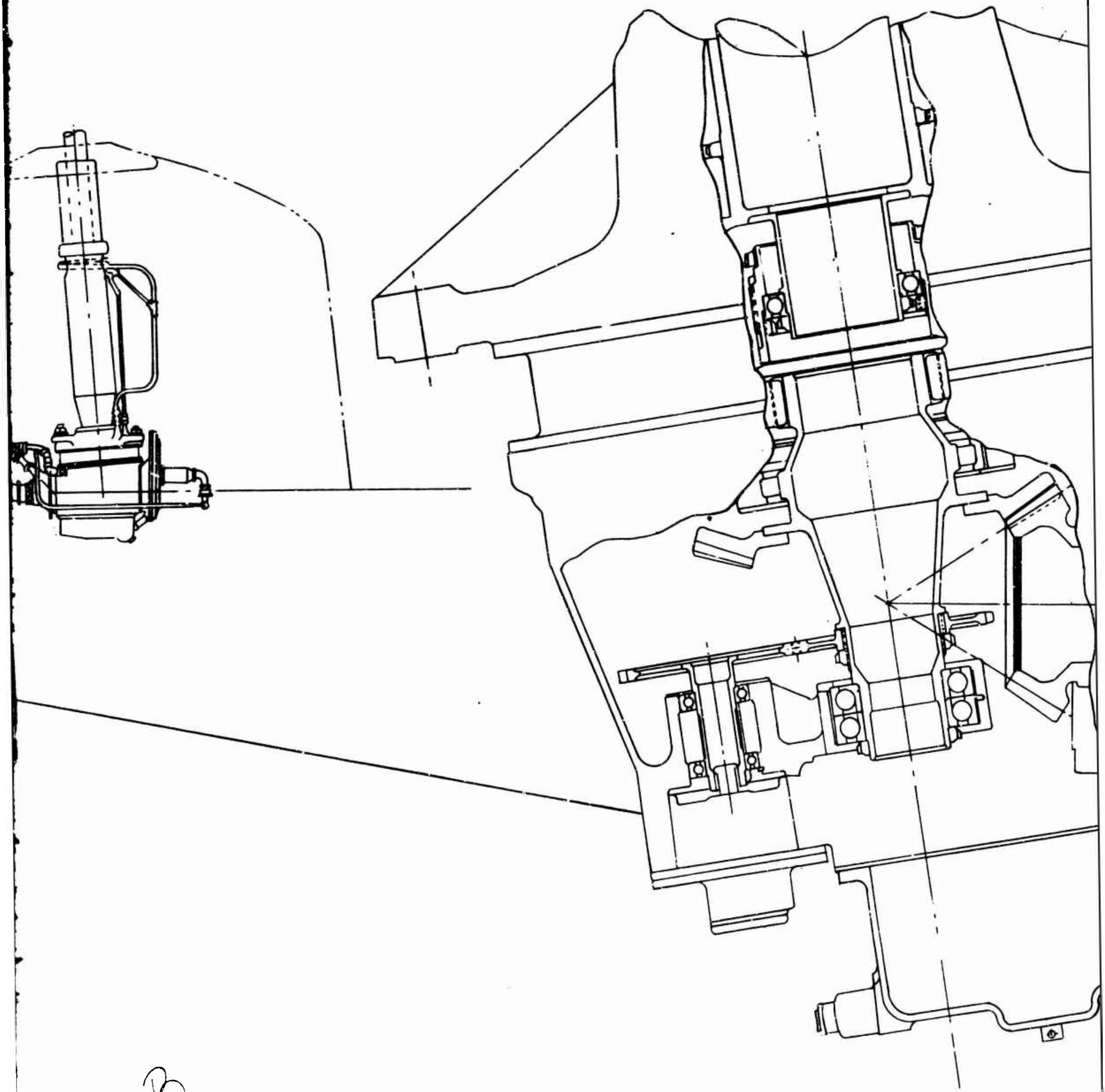


Figure 18. CH-47A Transmission System.

A



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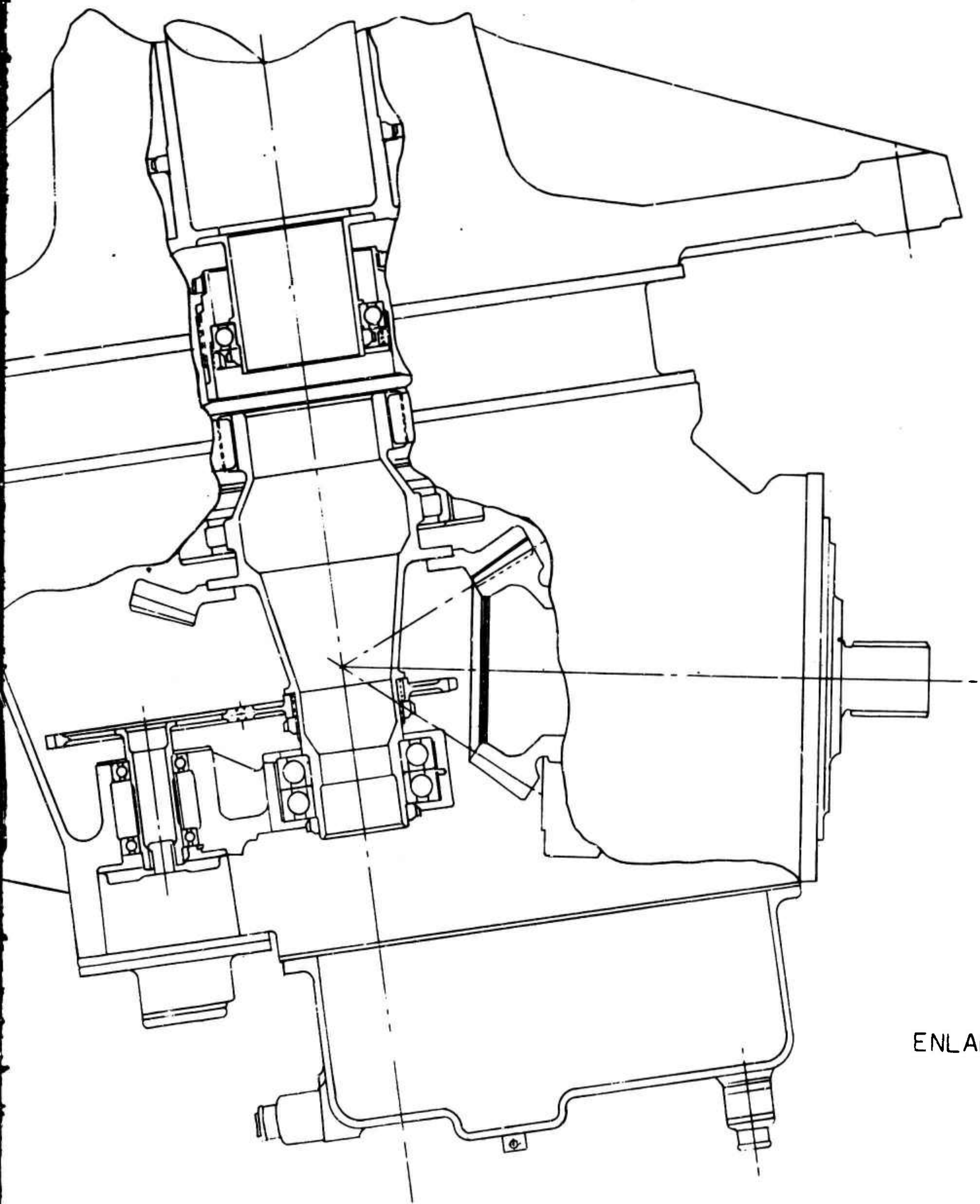

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Lubrication and Cooling System Description

Each transmission has its own lubrication circuit. A multielement pump for the engine transmissions and combining transmission is housed in the combining transmission. A separate three-section oil cooler is provided in the pylon to cool these transmissions. The forward main transmission oil pump is located in the lower portion of the transmission assembly and is driven through a spur gear mounted to the lower driven bevel shaft. Oil from the sump is pumped aft through a long oil line parallel and adjacent to the forward transmission drive shafting. The oil then passes through a separate oil cooler mounted in the pylon. From the oil cooler, the oil again flows through a long oil line adjacent to the drive shaft to the forward transmission oil inlet manifold. The aft main transmission oil pump mounted in, and driven by, the accessory gearbox circulates oil from the sump through another separate oil cooler located in the pylon and finally delivers oil to the inlet manifold. Cooling air is drawn through the three separate oil coolers in the pylon by a single blower, shaft driven from the aft main transmission. The transmission system lubrication and cooling schematic is shown in Figure 19.

Design Data

Two shaft turbine engines power the CH-47A helicopter. At military rated power, the earlier T-55-L-5 engine develops 2200 SHP and the later T-55-L-7 engine develops 2650 SHP. Because of limitations in the drive system, 2480 HP per engine is transmitted through the transmission system to the rotors. Since the forward transmission has long inlet and return oil lines to the aft-mounted oil cooler, it was considered by the investigators to be the most vulnerable portion of the CH-47A transmission system. This component therefore forms the basis of the adaptability study. A similar approach will also apply to the aft main transmission. A summary of the available cooling and lubrication basic design data is included as Table XIX.

Rotor Shaft Cooling System

System Description

An extremely invulnerable oil system is achieved by locating a cylindrical oil-to-air heat exchanger within the main shaft of the CH-47A forward main transmission. In this design the heat exchanger is stationary and is supported by the lower shaft bearing support on the bottom and by a sealed bearing on the top. The inlet and outlet oil lines are provided within the air duct. A shaft-driven axial blower draws ambient cooling air through the system and exhausts through an airframe-mounted exit duct. The system is shown in Figure 20.

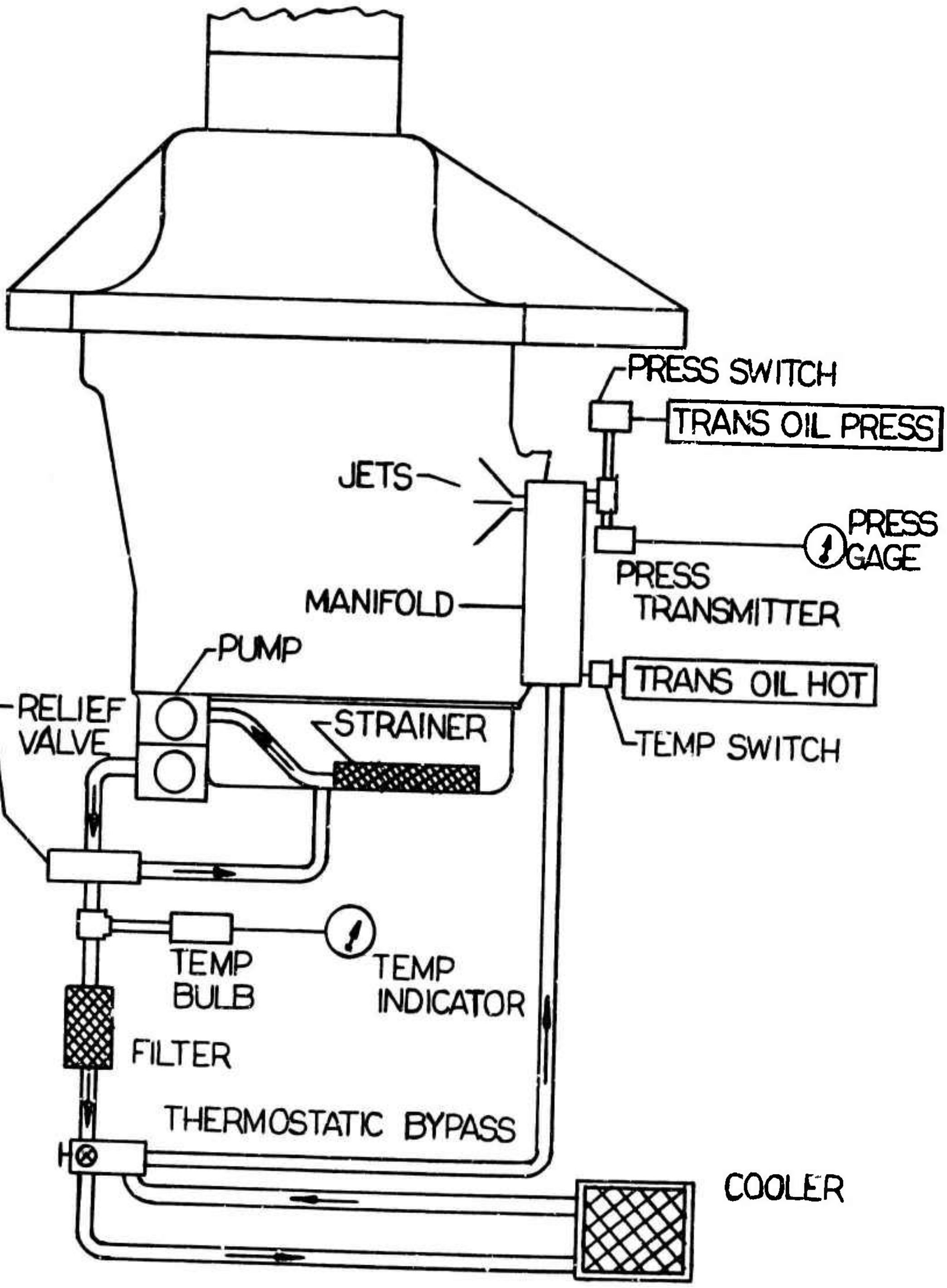


Figure 19. CH-47A Lubrication System Schematic.

TABLE XIX. COOLING AND LUBRICATION SYSTEM BASIC DATA - CH-47A FORWARD TRANSMISSION	
Parameter	Value
Maximum Continuous Input Power (HP)	2,640
Rotor Torque (ft-lb)	71,300
Input Speed (rpm)	7,175
Output Speed (rpm)	230
Oil Mass Flow Rate (gpm)	17.2
Total Forward Transmission Heat Rejection Rate (Btu/min)	2,020

Heat Exchanger

A shell and tube-type heat exchanger with oil directed through the shell side over the tubes by segmented baffles and air drawn through the tubes is employed. The core is located within the driven bevel gear lower shaft which rotates on the same center line as the upper main shaft. The lower shaft has a larger bore and permits the use of a larger core for less air-side pressure restriction. A welded shell extension provides rigidity for the core, so that simple supports at the top and bottom of the heat exchanger are all that are required to secure the installation. During buildup of the gearbox, the upper housing, upper main shaft, and bearing support assembly are installed; then the heat exchanger is fitted into the upper shaft. A slight modification to the first-stage planetary support ball bearing assembly is required and is shown in Figure 20. The planetaries and bevel-gear-driven shaft are then installed around the heat exchanger. When the driven bevel lower thrust bearings are installed, the lower heat exchanger support is installed. Oil-side connections to the heat exchanger are made via oil transfer tubes and O-rings that complete the oil circuit between the sump plumbing and the heat exchanger.

The heat exchanger is fabricated in its entirety from 321 series stainless steel for strength and corrosion resistance. High shell strength is required to ensure that buckling or high deflections do not occur and to permit the heat exchanger to contact the gearbox shafting. Stainless steel is also readily dip-brazed and will facilitate the

fabrication of oil-tight assemblies. The shell has an outside diameter of 4.44 inches and is made from .032-inch-thick stock. The core length is 1.5 feet including inlet and outlet oil headers. The shell contains 684 stainless steel air tubes closely packed in an equilateral staggered pattern .150 inch between centers. The air tube outside diameter is .125 inch. The air tube inside diameter is .115 inch. The outlet oil line, which passes through the core from the upper header, and the inlet oil line, which joins the lower header, occupy space in the air-side passage and displace some air tubes in the core. The resulting loss in air-side area is much less severe than if the shell outside diameter were reduced to accommodate conventional external shell-side oil connections. The 684 air tube figure is calculated after the allowance for displacement of tubes by the oil lines is made. The oil-side baffling is of the segmented type on a 2-inch spacing pattern, resulting in nine baffles.

Axial Blower

A single-stage axial blower draws ambient air from the area above the center line of the forward rotor through the integral cooling system and exhausts heated air overboard through a discharge duct. The blower is shaft driven directly from the forward main transmission. The blower housing assembly, rotor vane, and stator vane assemblies are fabricated from aluminum. Sealed, prepacked bearings support the rotating portion of the blower assembly.

Blower Drive

The axial blower is driven by a compact spur gear and bevel gear takeoff assembly added to the lower portion of the forward main transmission. The face width of the oil pump drive spur gear splined to the driven main bevel lower shaft is increased to accommodate the blower drive takeoff power requirement. The blower drive gearing increases speed from the driven bevel lower shaft speed to blower drive shaft speed of 9000 rpm. The blower drive shaft incorporates flexible disc couplings at both ends.

Ducting and Accessories

The transition from the lower portion of the rotor shaft heat exchanger to the inlet section of the draw-through blower is a structural portion of the oil sump. The oil sump is redesigned to accommodate the transition and the oil line transfer tubes from the existing plumbing to the heat exchanger. The basic characteristics of the sump as an oil reservoir are unchanged. The thermostatic bypass valve assembly required to regulate gearbox operating temperature is installed in the

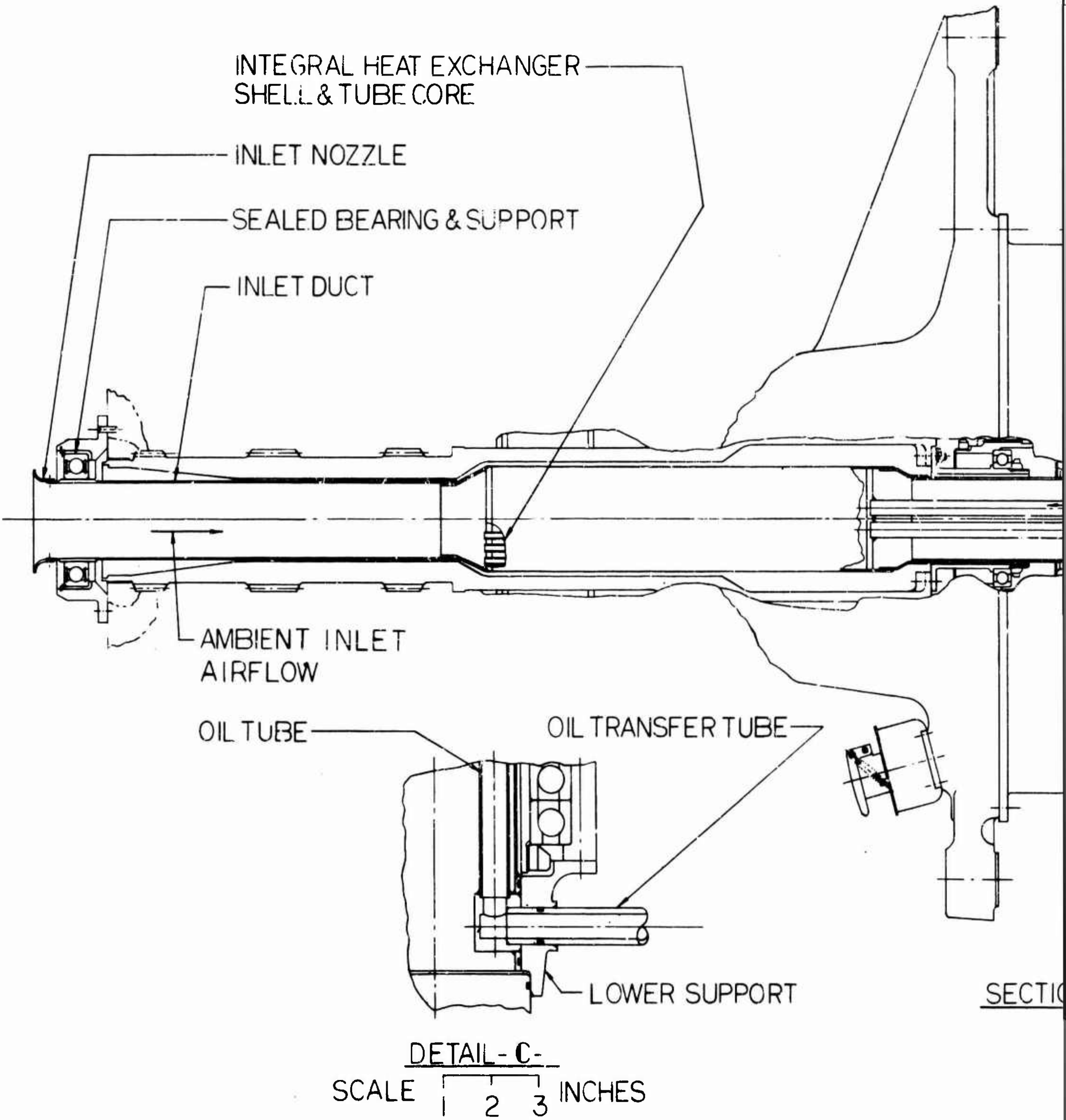
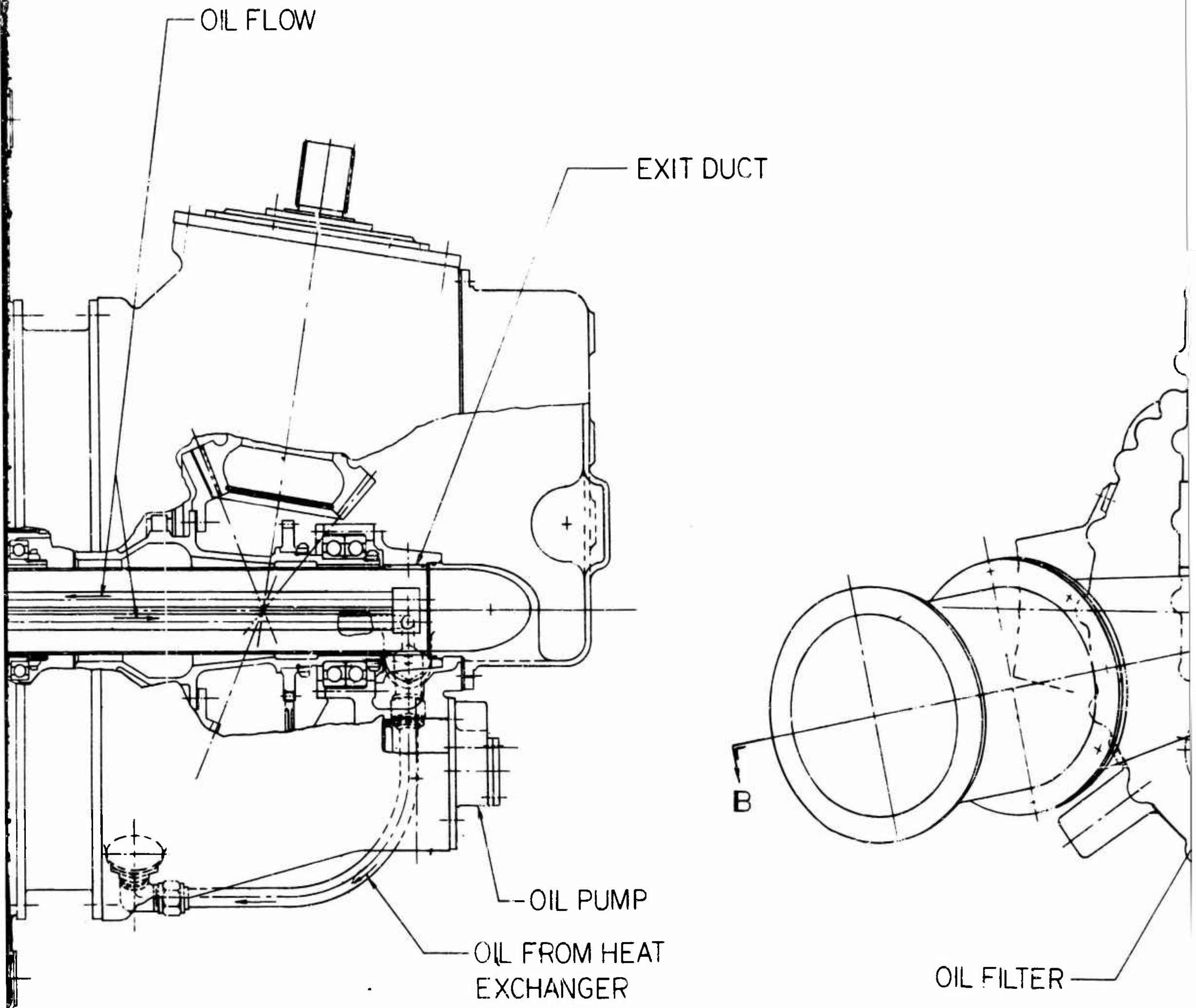


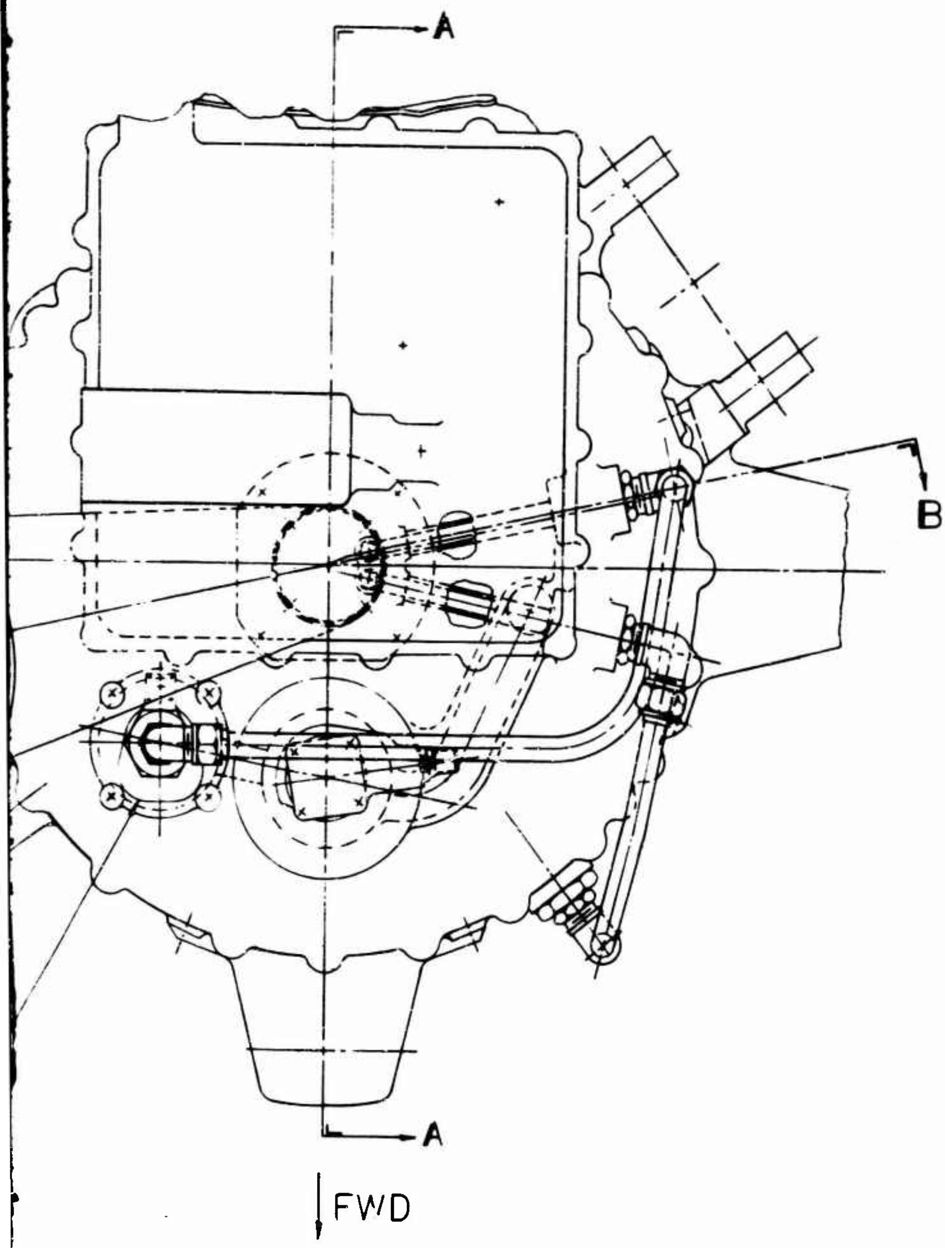
Figure 20. CH-47A Rotor Shaft Cooling System.

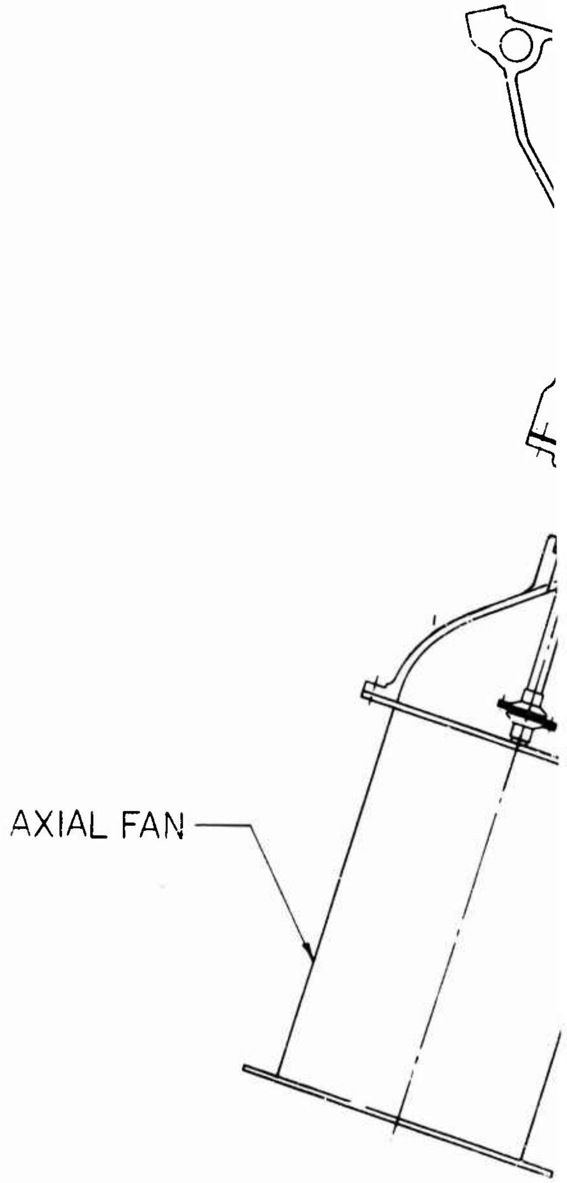
A


OIL FLOW
EXIT DUCT
B
OIL PUMP
OIL FROM HEAT EXCHANGER
OIL FILTER

SECTION -A-A-

B





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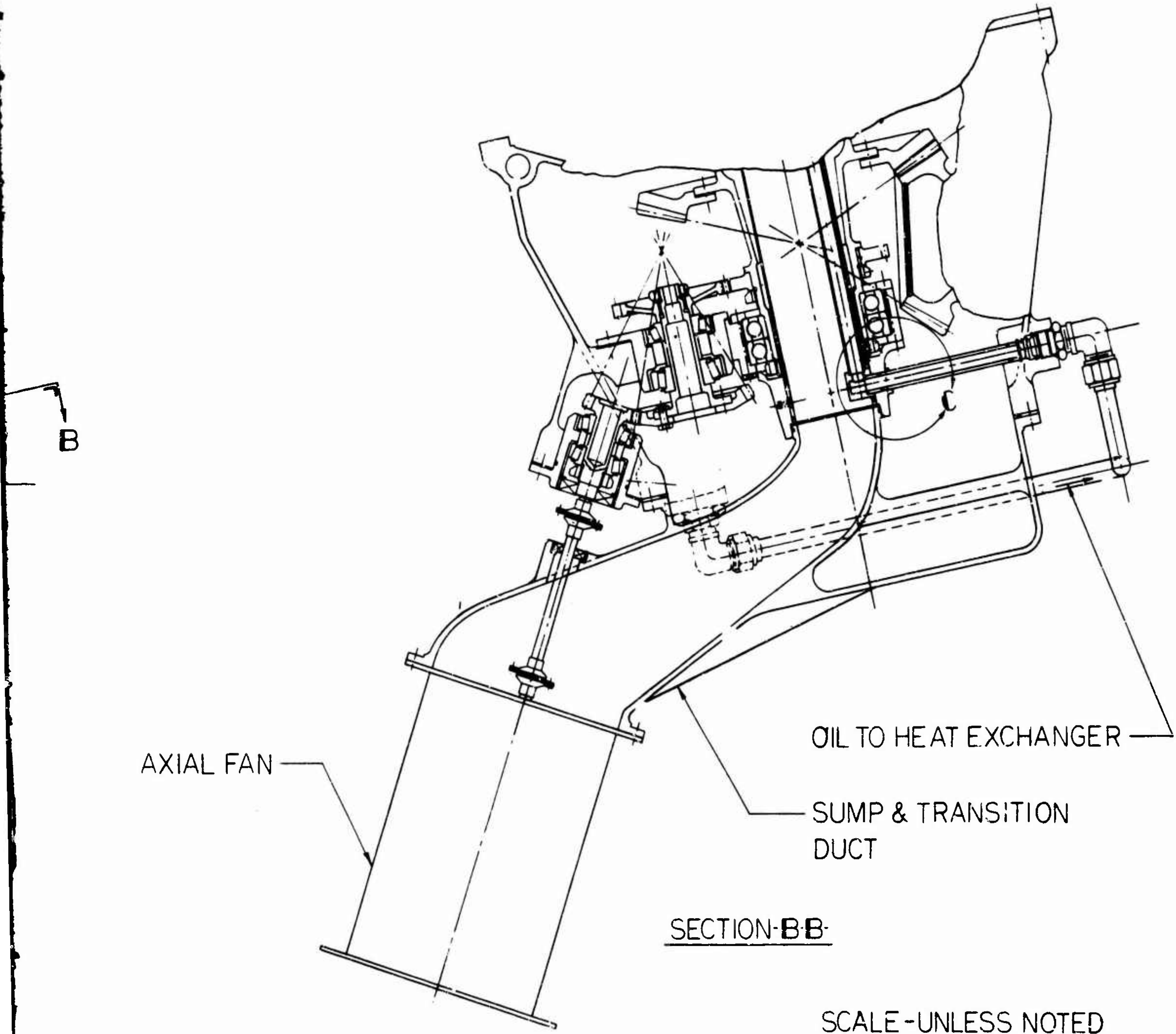

SECTION-B-B-

SCALE-UNLESS NOTED

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sump adjacent to the oil transfer tube fitting bosses. In cold weather operation, circulation of oil to the heat exchanger is restricted by bypassing some oil directly to the oil outlet line in the bypass valve assembly.

Performance Analysis

The analysis of the shell and tube-type heat exchanger within the rotor shaft for the CH-47A is similar to that for the CH-53A contained in Appendix I. Significant design parameters are presented in Table XX. The calculated weights for the component parts of the rotor shaft cooling system are summarized in Table XXI.

Oil Sump Cooling System

System Description

For this system, a compact finned-plate heat exchanger is mounted external to, but integral with, the gearbox oil sump. The heat exchanger is provided with inlet and outlet oil-side manifolds to which the existing portion of the oil circuit is connected. Air is passed through the core by an axial blower driven by the forward main transmission. The oil sump cooling system installation is shown in Figure 21.

Heat Exchanger

The finned-plate exchanger is fabricated entirely from aluminum fabricated by dip-brazing. The core is built up from alternate oil and air passages separated by .020-inch flat plates. Zig-zag foil fins are employed on the air side to facilitate heat transfer. The outer shell plates are .060 inch thick. There are eight air passages .405 inch wide and seven oil passages .143 inch wide. Length of the core is calculated to be 1.1 feet. Frontal dimensions of the core are 4.5 inches by 7.2 inches.

Axial Blower

A single-stage axial blower very similar to the unit described for the CH-54A oil sump cooling system is employed. The blower is sized for the "pusher" mode. A draw-through system can be used also. The shaft horsepower requirement increases slightly if the draw-through system is selected.

TABLE XX. PERFORMANCE PARAMETERS - CH-47A ROTOR SHAFT COOLING SYSTEM

Parameter	Value
Oil Temperature Into Heat Exchanger (°F)	280
Oil Temperature Into Gearbox (°F)	256
Ambient Air Temperature, Maximum (°F)	104
Air Temperature out of Heat Exchanger (°F)	262
Heat Exchanger Heat Load (Btu/min)	1,560
Heat Rejected Through Housings (Btu/min)	460
Air Mass Flow Rate (lb/hr)	2,460
Oil Mass Flow Rate (gpm)	17.2
Heat Exchanger Effectiveness	.9
Air-Side Heat Transfer Area (ft^2)	31.3
Oil-Side Heat Transfer Area (ft^2)	35.8
Cross-Sectional Air-Side Area (ft^2)	.045
Air Mass Velocity (lb/hr ft^2)	54,500
Air-Side Reynolds Number	9,600
Air-Side Film Heat Transfer Coefficient (Btu/hr ft^2 °F)	51.7
Air Velocity Through Core (ft/sec)	320
Oil Mass Velocity (lb/hr ft^2)	1,040,000
Oil-Side Reynolds Number	1,130
Oil-Side Film Heat Transfer Coefficient (Btu/hr ft^2 °F)	460
Overall Heat Transfer Coefficient (Btu/hr ft^2 °F)	46.5
Calculated Heat Exchanger Core Length (ft)	1.5
Total Air-Side Suction Heat (in. of H_2O)	72.8
Blower Air Horsepower (HP)	10.3
Blower Shaft Horsepower (HP)	14.8
Oil-Side Pressure Drop (psi)	28.8

TABLE XXI. WEIGHT SUMMARY - CH-47A ROTOR SHAFT COOLING SYSTEM	
Cooling System Component	Weight (lb)
Heat Exchanger, Stainless Steel	13.52
Heat Exchanger Support Structure	4.3
Air-Side Ducting	8.2
Oil-Side Plumbing and Accessories	4.6
Blower	20.1
Blower Drive and Shafting	12.0
Total System Dry Weight	62.72

Blower Drive

A compact step-up geared takeoff similar to the assembly described for the CH-47A rotor shaft cooling system is used.

Ducting and Accessories

The transition duct is a structural duct providing a smooth transformation from the blower discharge to the rectangular cross-sectional inlet to the heat exchanger. The transition duct provides the support structure for the blower assembly. A thermostatic bypass device for cold weather operation is provided as part of the heat exchanger oil outlet manifold.

Performance Analysis

The calculated weights of the component parts of the oil sump cooling system are presented in Table XXII. The results of the design analysis of the finned-plate heat transfer system are summarized in Table XXIII. The procedure is similar to that for the main shaft heat exchanger analysis of Appendix I and therefore is not repeated.

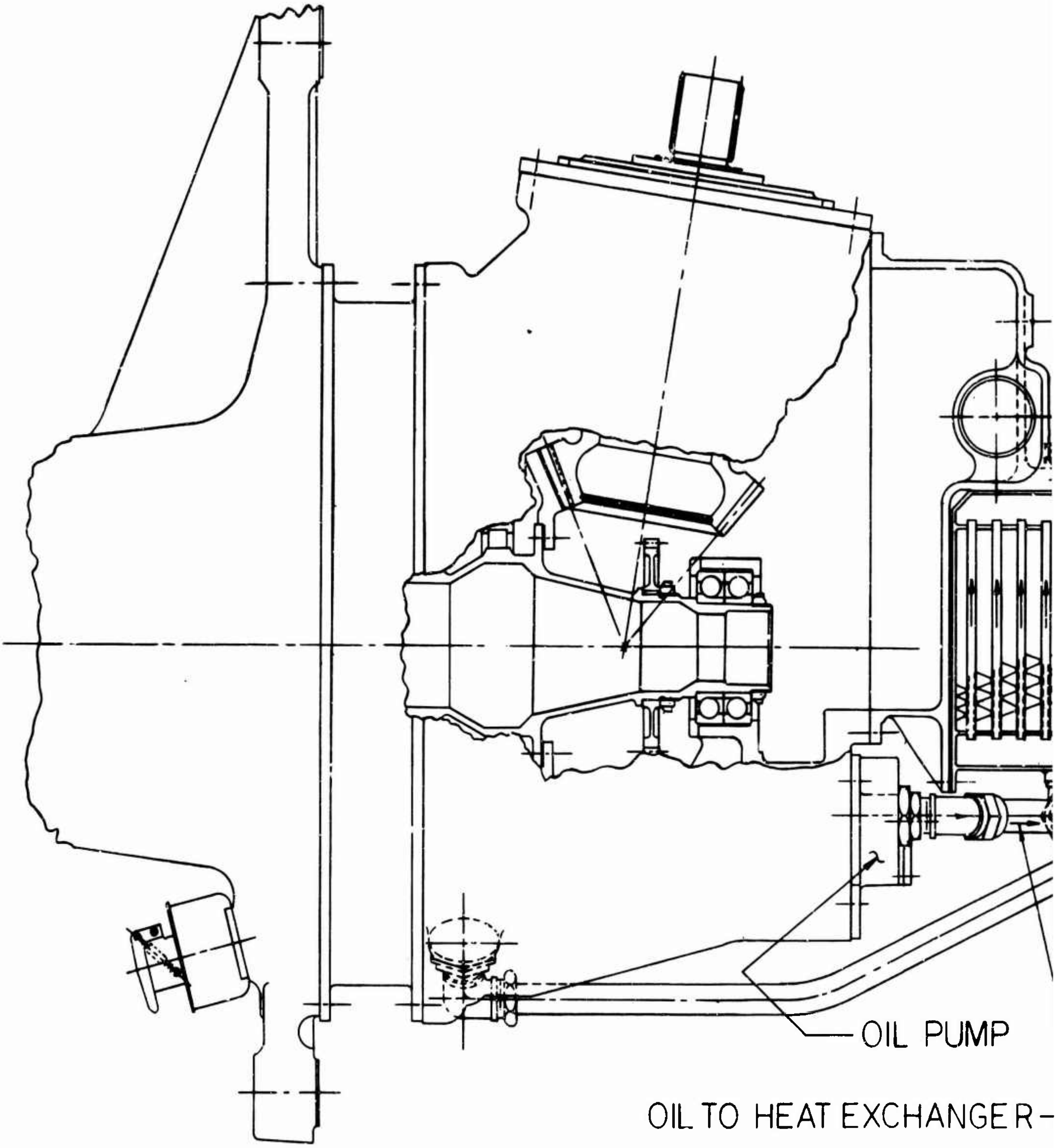
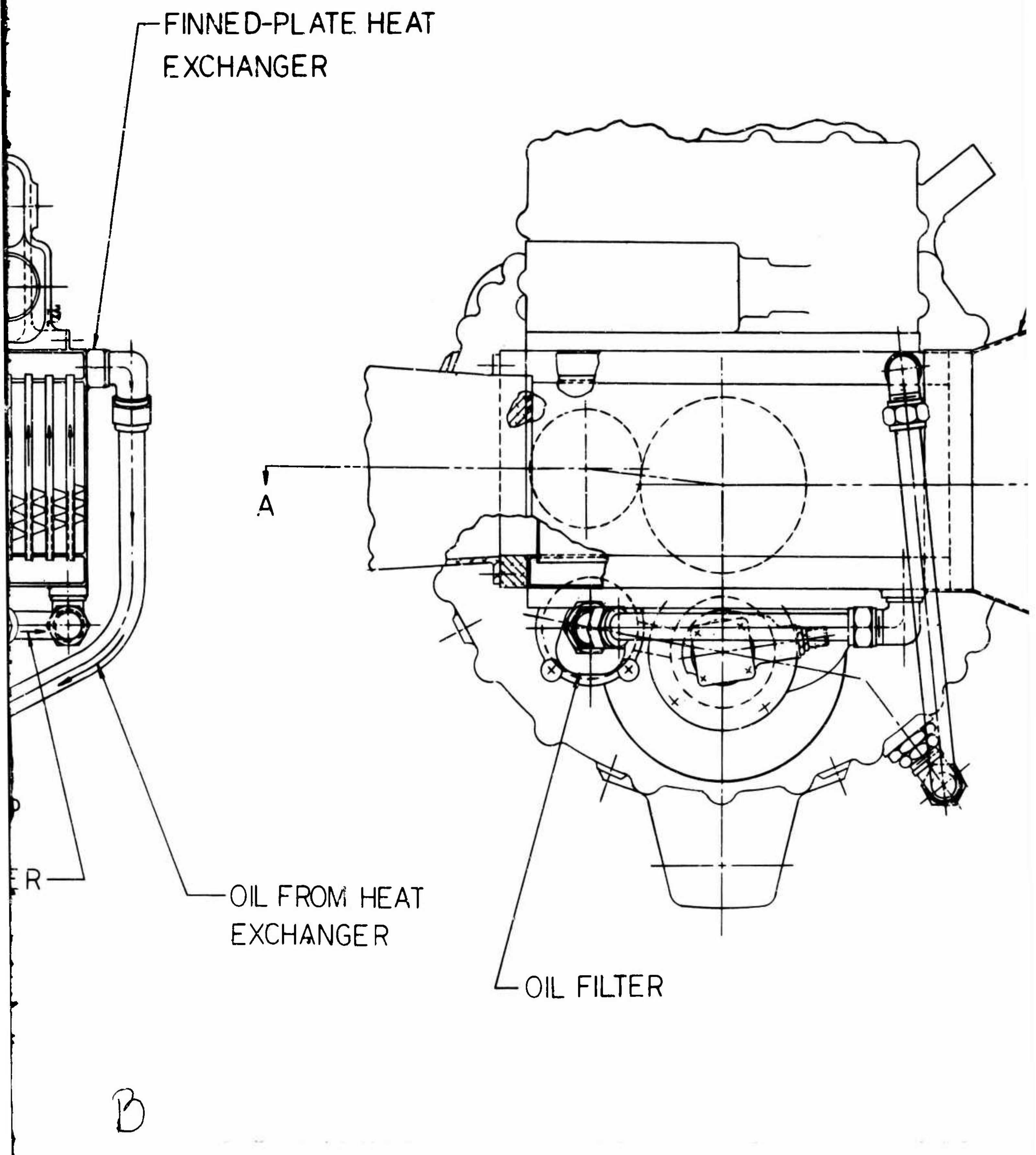
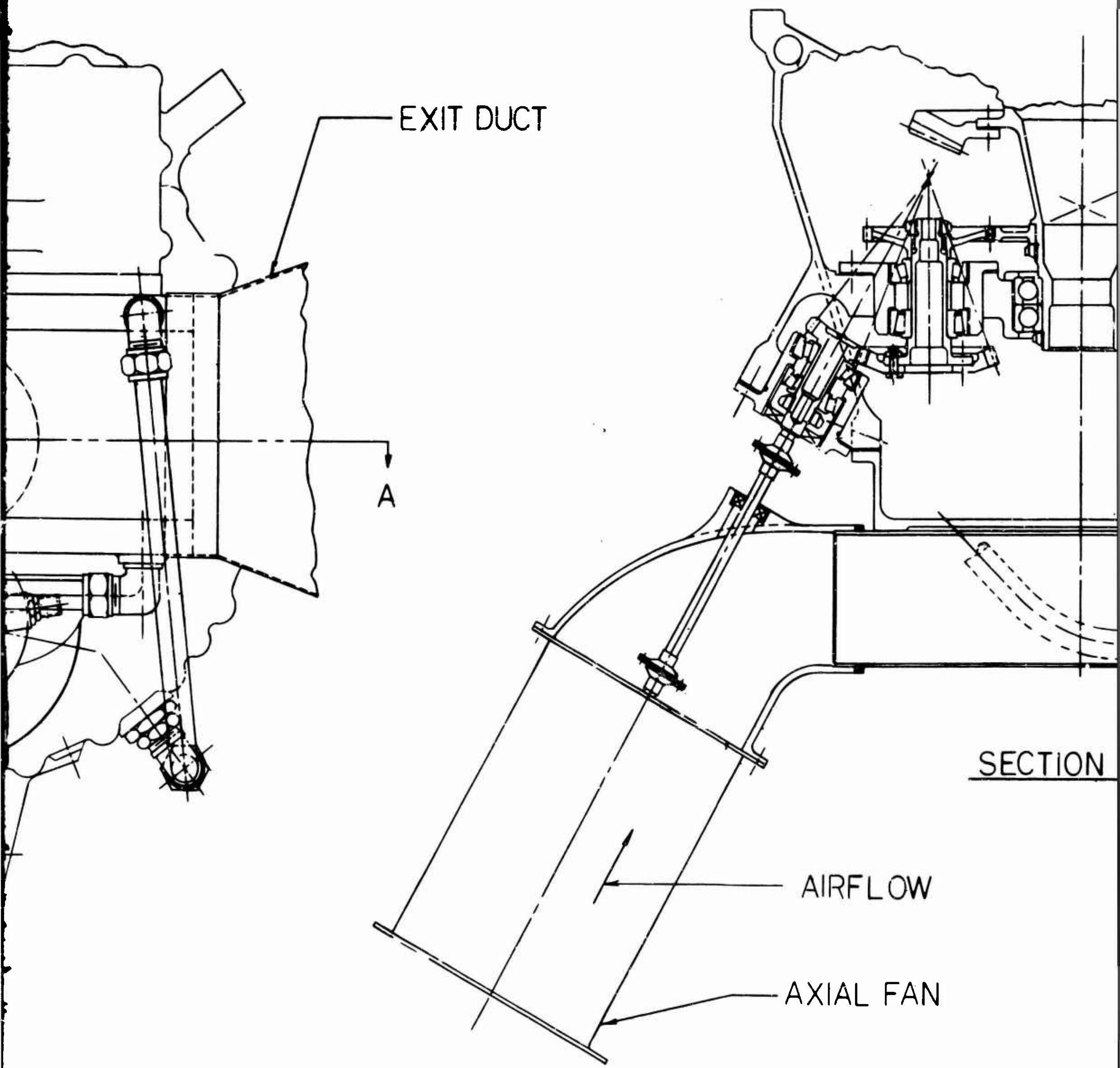


Figure 21. CH-47A Oil Sump Cooling System.

A


FINNED-PLATE HEAT EXCHANGER
A
ER
OIL FROM HEAT EXCHANGER
OIL FILTER
B



EXIT DUCT
A
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AIRFLOW
AXIAL FAN

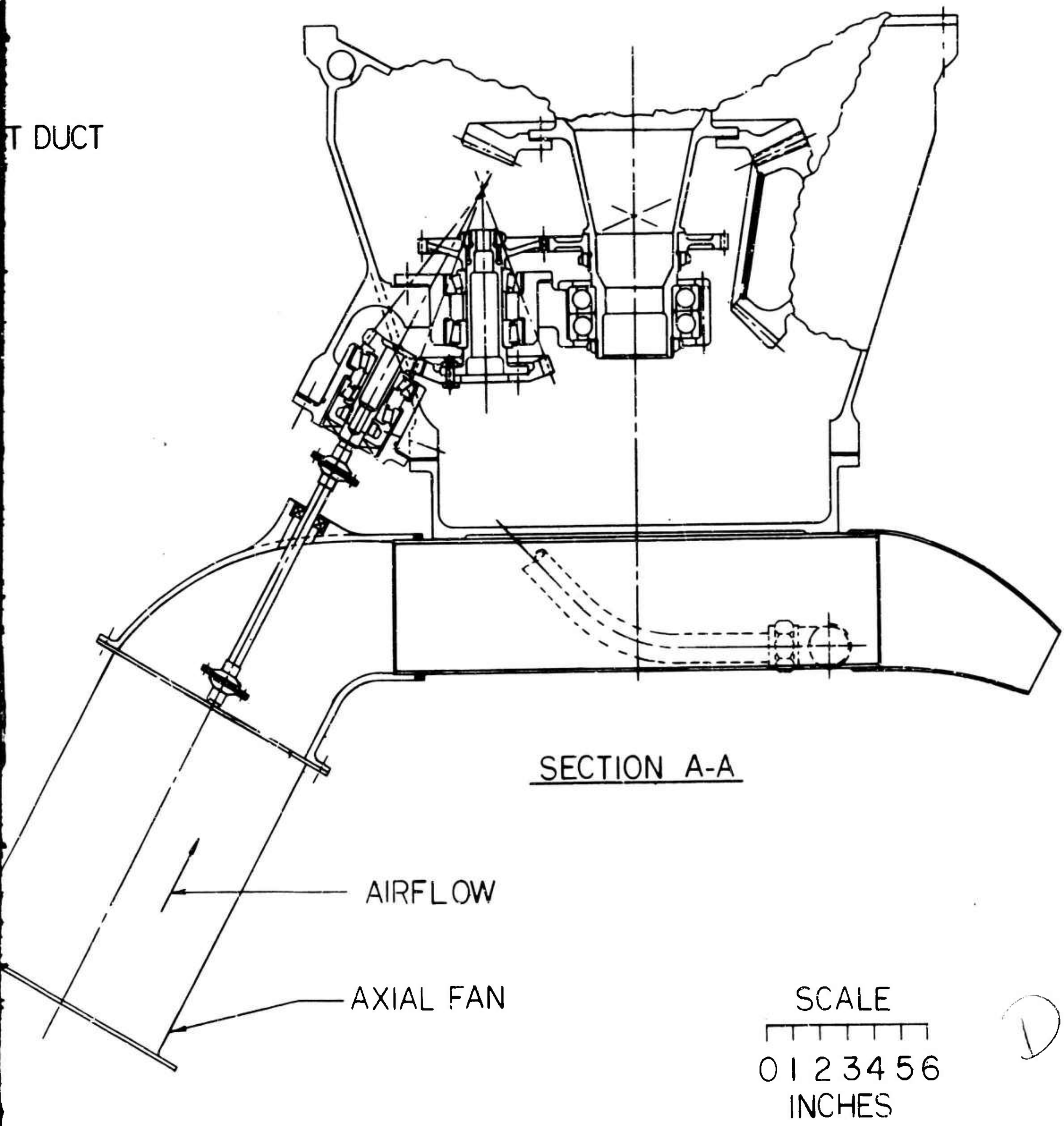

T DUCT
SECTION A-A
AIRFLOW
AXIAL FAN
SCALE
0 1 2 3 4 5 6
INCHES

TABLE XXII. WEIGHT SUMMARY - CH-47A OIL SUMP COOLING SYSTEM	
Cooling System Component	Weight (lb)
Heat Exchanger, Aluminum	11.3
Heat Exchanger Support Structure	3.8
Air-Side Ducting	6.8
Oil-Side Plumbing and Accessories	4.0
Blower	20.0
Blower Drive and Shafting	11.6
Total System Dry Weight	57.5

TABLE XXIII. PERFORMANCE PARAMETERS - CH-47A OIL SUMP COOLING SYSTEM	
Parameter	Value
Oil Temperature Into Exchanger (^{o}F)	280
Oil Temperature Into Gearbox (^{o}F)	256
Ambient Air Temperature, Maximum (^{o}F)	104
Adiabatic Heat of Compression Across Blower (^{o}F)	13
Actual Air Temperature Into Heat Exchanger (^{o}F)	117
Heat Exchanger Oil Heat Load (Btu/min)	1,560
Air Mass Flow Rate (lb/hr)	5,010
Oil Mass Flow Rate (gpm)	17.2
Heat Exchanger Effectiveness	.5
Air-Side Heat Transfer Area (ft^2)	32.8

TABLE XXIII - Continued	
Parameter	Value
Oil-Side Heat Transfer Area (ft^2)	8.4
Cross-Sectional Air-Side Area (ft^2)	.14
Air Mass Velocity (lb/hr ft^2)	36,500
Air-Side Reynolds Number	12,800
Air-Side Film Heat Transfer Coefficient (Btu/hr ft^2 ^oF)	34.6
Air Velocity Through Core (ft/sec)	160
Oil Mass Velocity (lb/hr ft^2)	31,000
Oil-Side Reynolds Number	90
Oil-Side Film Heat Transfer Coefficient (Btu/hr ft^2 ^oF)	77.5
Overall Heat Transfer Coefficient (Btu/hr ft^2 ^oF)	30.8
Calculated Heat Exchanger Core Length (ft)	1.1
Total Air-Side Pressure Head (in. of H_2O)	22.0
Blower Air Horsepower (HP)	4.6
Blower Shaft Horsepower (HP)	6.6
Oil-Side Pressure Drop (psi)	6.1

UH-1D HELICOPTER

Basic Data

Vehicle Description

The UH-1D is a single-rotor helicopter employing a two-bladed semirigid main rotor assembly and an antitorque two-bladed semirigid tail rotor assembly. A single shaft turbine engine drives the rotor system.

Drive Train Configuration

The main transmission is mounted forward of the engine and is coupled to the power turbine by a short drive shaft. A tail rotor takeoff drive from the aft portion of the main transmission provides torque to drive the tail rotor, which is transmitted through a 42° intermediate gearbox and terminates in a 90° tail rotor gearbox. The main transmission primary drive is accomplished via an input freewheel unit, an input spiral bevel mesh, and a two-stage planetary.

Lubrication and Cooling System Description

Main transmission lubrication and cooling are accomplished by a self-contained pressure-fed oil system. An oil pump is provided in the wet sump and is shaft driven internally from the tail takeoff drive of the main transmission. Oil from the sump is circulated to the transmission oil cooler, which is attached to the lower portion of the engine oil cooler. Cooled oil is then returned to the distribution system of the gearbox. Cooling air for the transmission oil cooler and the engine oil cooler is provided by a turbine fan driven from engine compressor section bleed air. The UH-1D transmission system lubrication schematic is shown in Figure 22, and the basic transmission schematic is depicted in Figure 23.

Design Data

The adaption of integral cooling methods to the UH-1D main transmission system is investigated in this section. A summary of the available cooling and lubrication basic design data is included in Table XXIV.

Rotor Shaft Cooling System

An integral cooling system utilizing a cylindrical heat exchanger mounted within the main rotor shaft is readily adapted to the UH-1D main transmission. The system is shown in Figure 24. Since the design of the individual components is very similar to that proposed for the CH-54A and CH-47A, a detailed description is not included. Tables XXV and XXVI summarize the performance and weight analyses conducted for this system.

Oil Sump Cooling System

An integral oil sump cooler was also investigated for the UH-1D main gearbox. In this system the heat exchanger, blower, and drive are very similar to those proposed for the other two aircraft of this report. The suggested installation is shown in Figure 25. Tables XXVII and XXVIII summarize the performance and weight analyses conducted for this system.

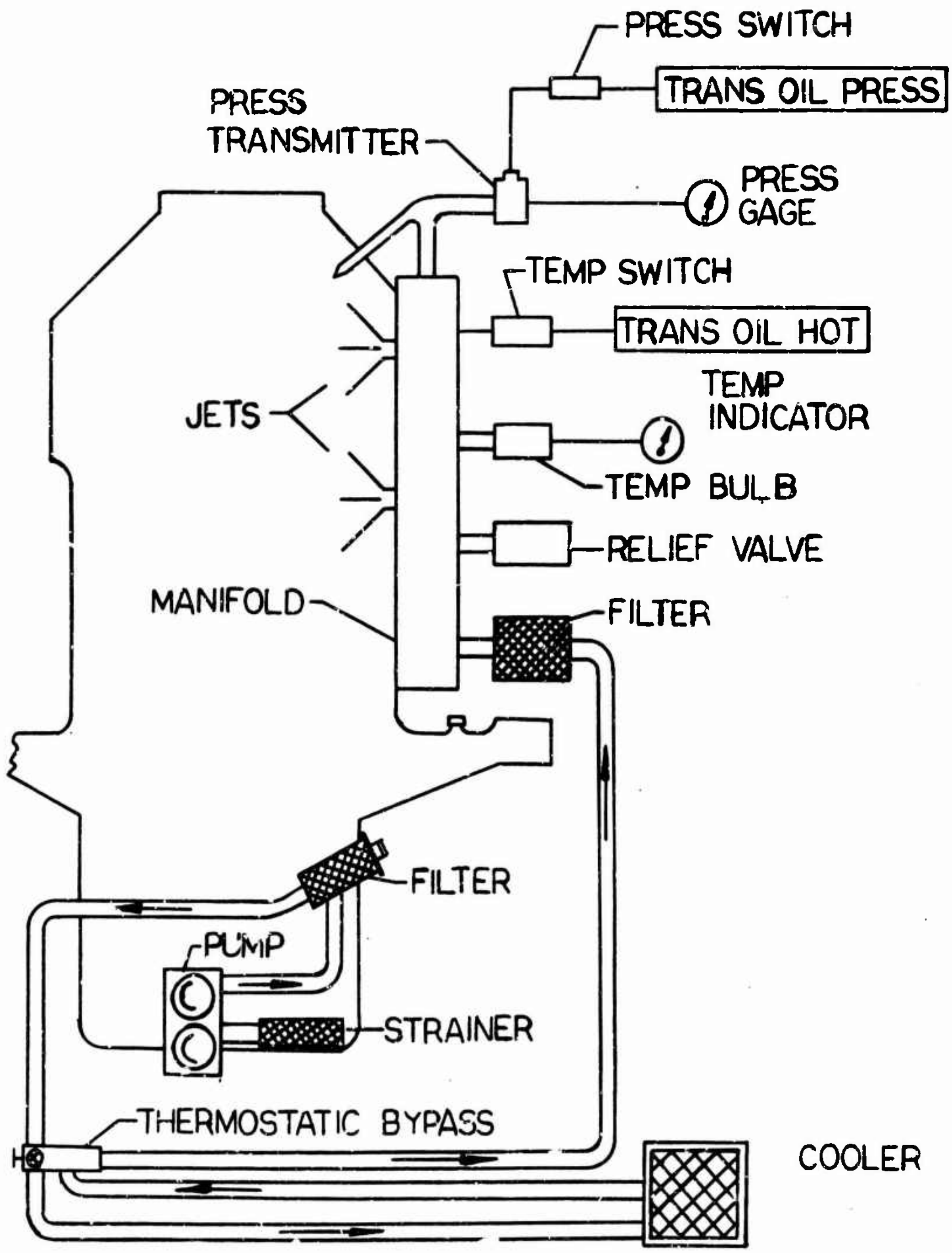


Figure 22. UH-1D Lubrication System Schematic.

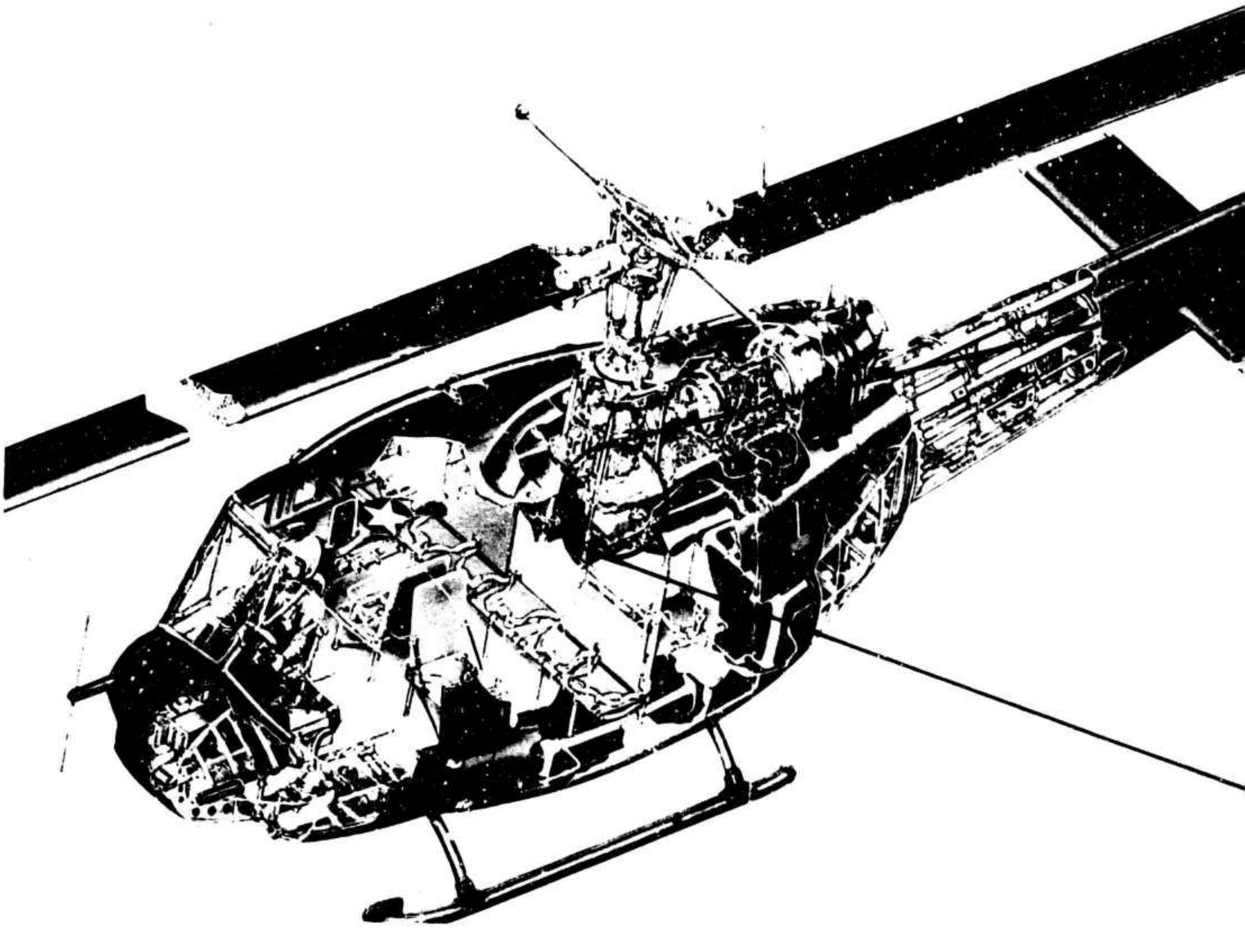
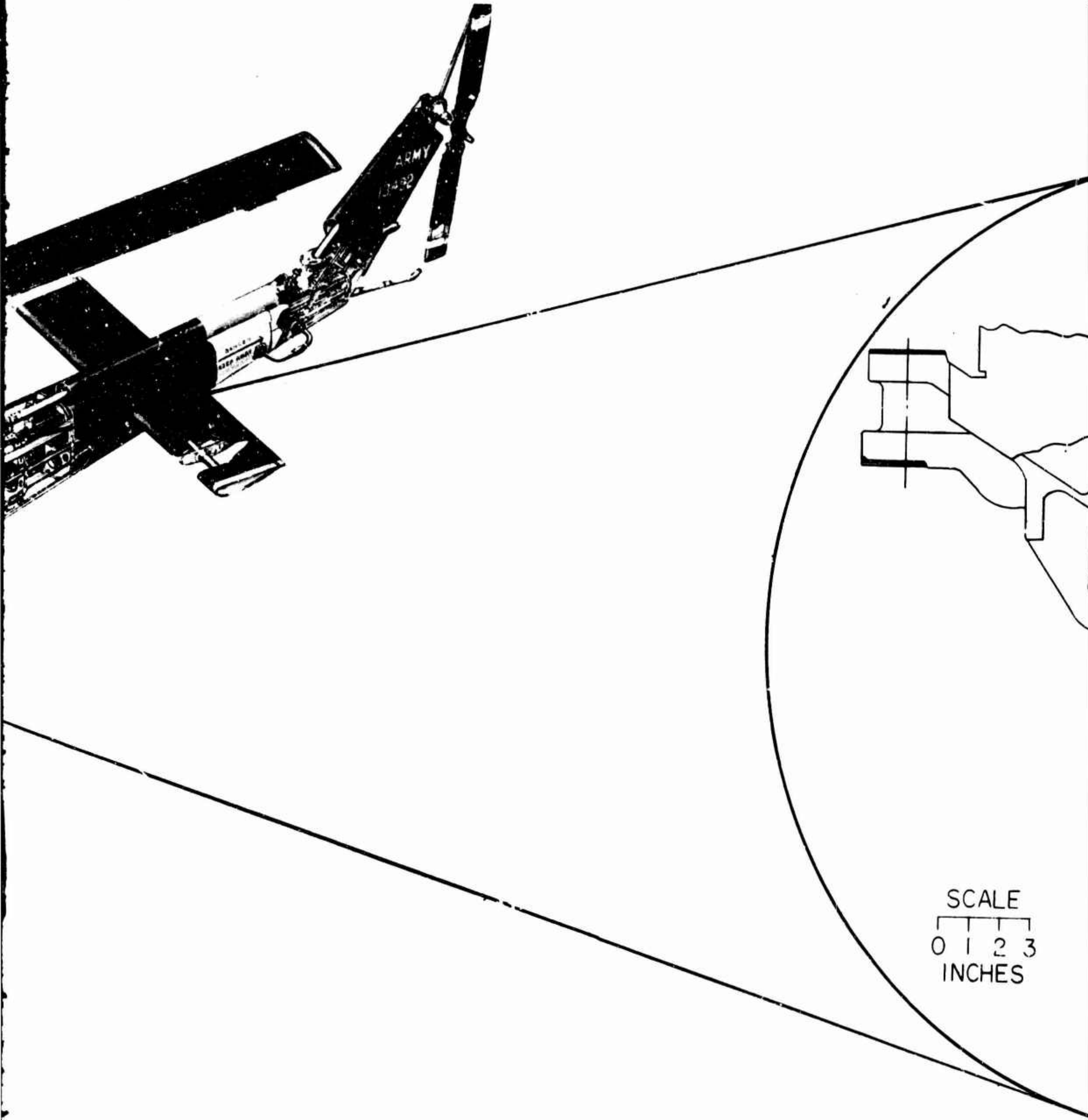
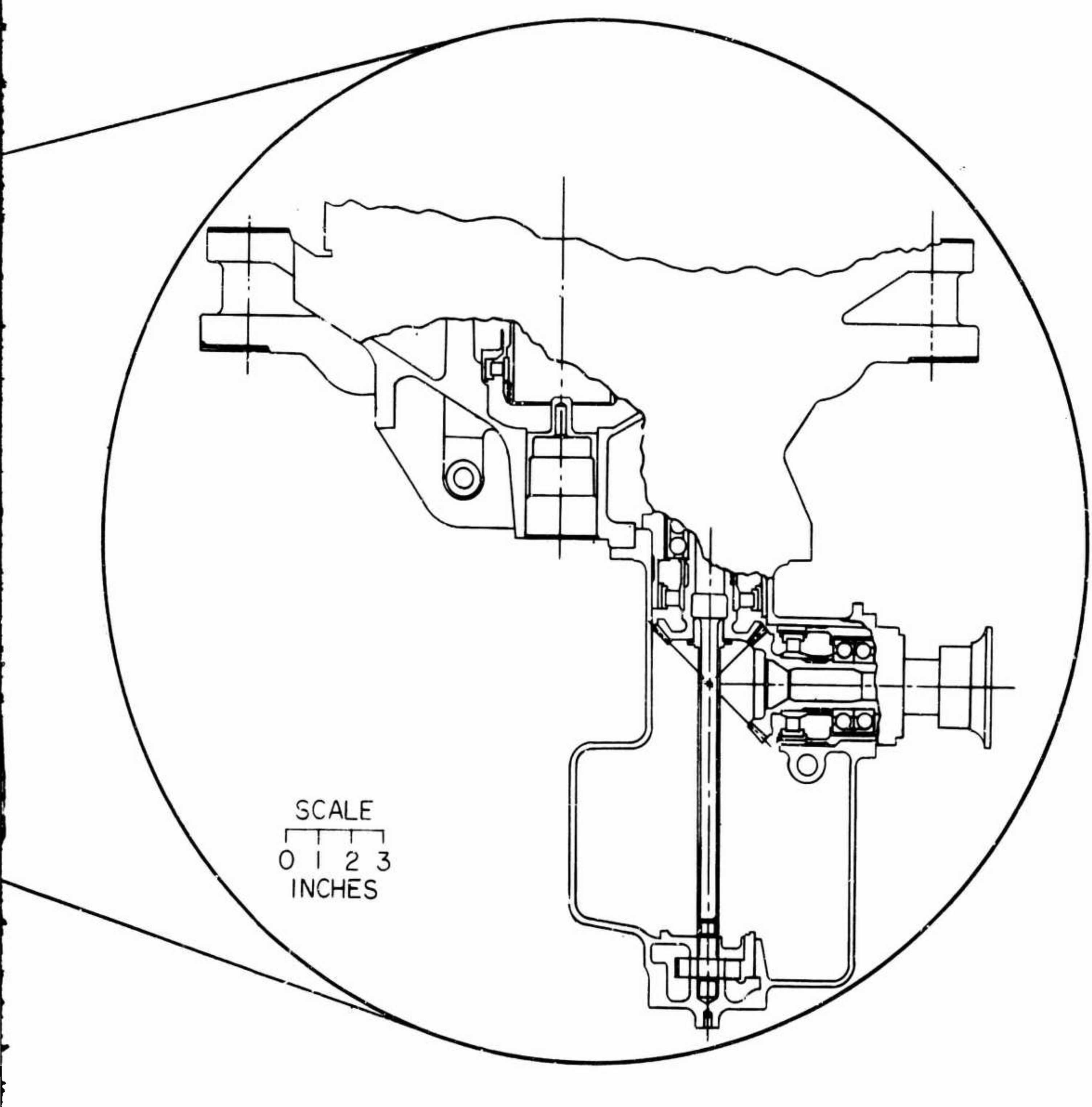


Figure 23. UH-1D Transmission System.

A



B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INCHES

TABLE XXIV. COOLING AND LUBRICATION SYSTEM BASIC DATA - UH-1D MAIN TRANSMISSION

Parameter	Value
Maximum Continuous Input Power (HP)	1,100
Rotor Torque (ft-lb)	17,824
Input Speed (rpm)	6,600
Output Speed (rpm)	324
Oil Mass Flow Rate (gpm)	12.5
Total Main Transmission Heat Rejection Rate (Btu/min)	933
Heat Rejection Rate Through Housings (Btu/min)	343

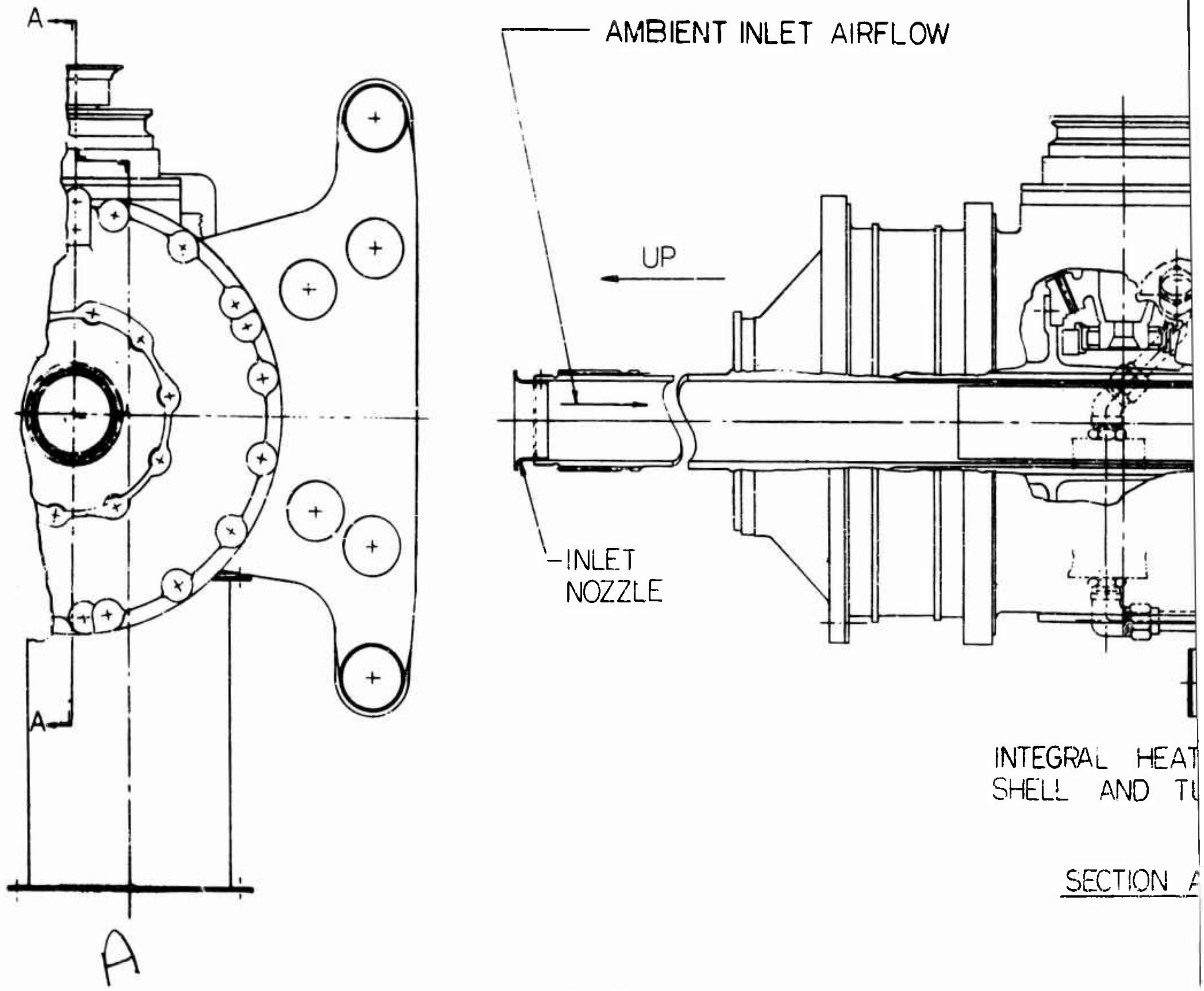
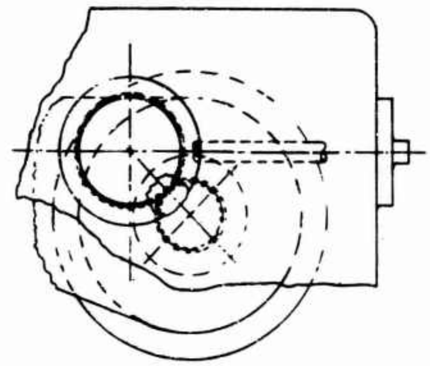
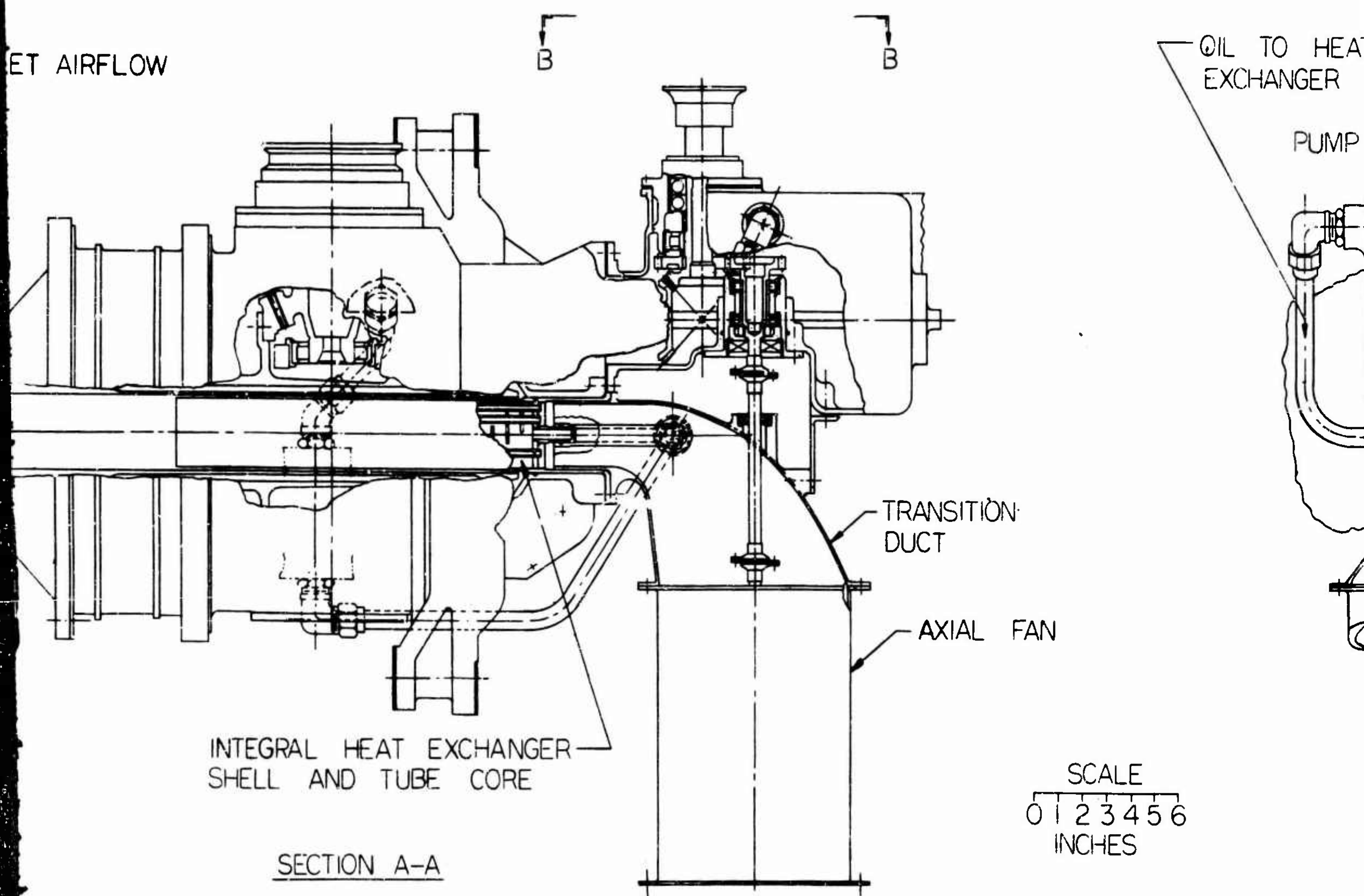


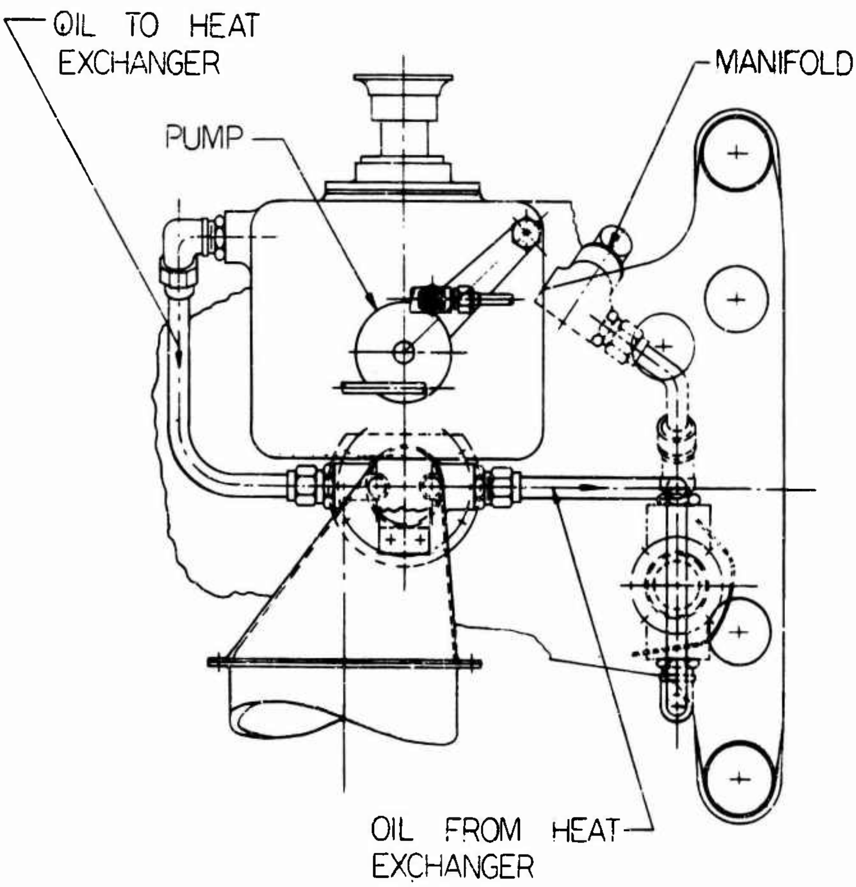
Figure 24. UH-1D Rotor Shaft Cooling System.



VIEW B-B



SECTION A-A



AN

SCALE
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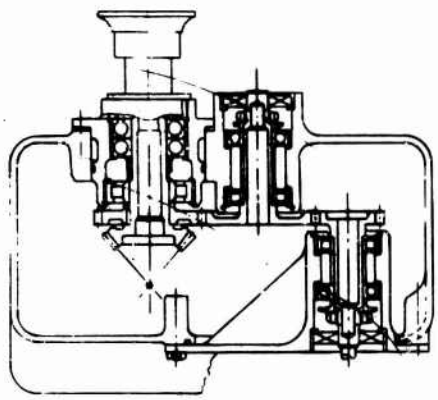
INCHES

TABLE XXV. PERFORMANCE PARAMETERS - UH-1D ROTOR SHAFT COOLING SYSTEM

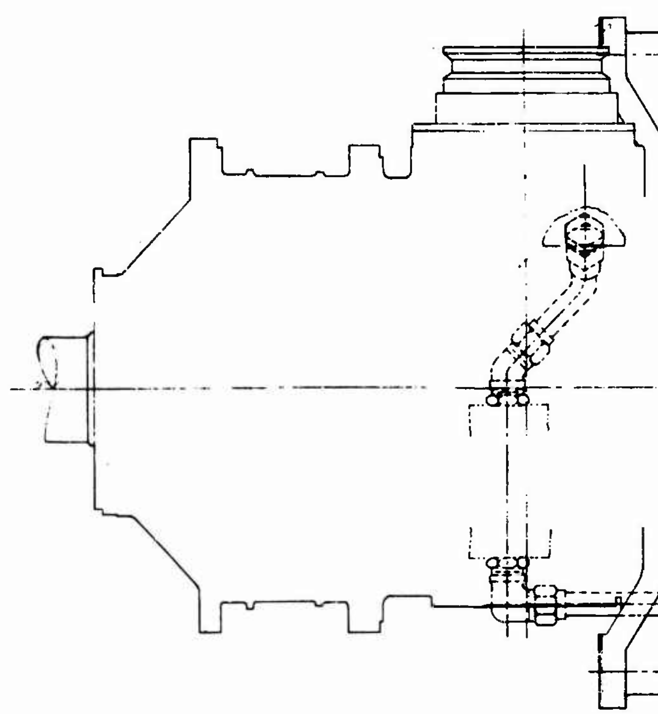
Parameter	Value
Oil Temperature Into Heat Exchanger (°F)	280
Oil Temperature Into Gearbox (°F)	267.5
Ambient Air Temperature, Maximum (°F)	104
Air Temperature out of Heat Exchanger (°F)	262
Heat Exchanger Heat Load (Btu/min)	590
Heat Rejected Through Housings (Btu/min)	343
Air Mass Flow Rate (lb/hr)	930
Oil Mass Flow Rate (gpm)	12.5
Heat Exchanger Effectiveness	.9
Air-Side Heat Transfer Area (ft^2)	12.5
Cross-Sectional Air-Side Area (ft^2)	.019
Oil-Side Heat Transfer Area (ft^2)	13.8
Air Mass Velocity (lb/hr ft^2)	49,700
Air-Side Reynolds Number	8,800
Air-Side Film Heat Transfer Coefficient (Btu/hr ft^2 °F)	48.6
Air Velocity Through Core (ft/sec)	290
Oil Mass Velocity (lb/hr ft^2)	1,030,000
Oil-Side Reynolds Number	1,090
Oil-Side Film Heat Transfer Coefficient (Btu/hr ft^2 °F)	456
Overall Heat Transfer Coefficient (Btu/hr ft^2 °F)	44
Calculated Heat Exchanger Core Length (ft)	1.5
Total Air-Side Suction Head (in. of H_2O)	58.3
Blower Air Horsepower (HP)	3.03
Blower Shaft Horsepower (HP)	4.35
Oil-Side Pressure Drop (psi)	15

TABLE XXVI. WEIGHT SUMMARY - UH-1D ROTOR SHAFT COOLING SYSTEM

Cooling System Component	Weight (lb)
Heat Exchanger, Stainless Steel	5.8
Heat Exchanger Support Structure	4.0
Air-Side Ducting	7.5
Oil-Side Plumbing and Accessories	4.2
Blower	15.0
Blower Drive and Shafting	12.0
Total System Dry Weight	48.5

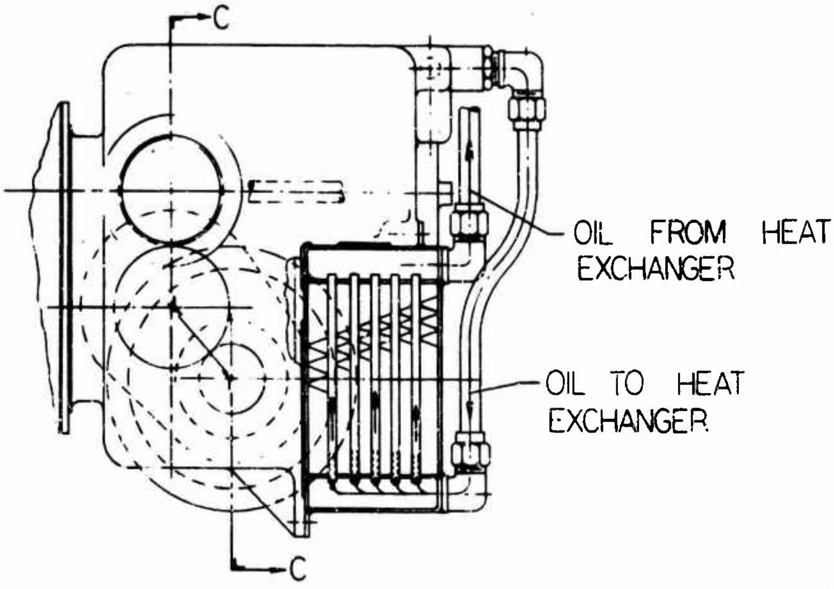


SECTION C-C

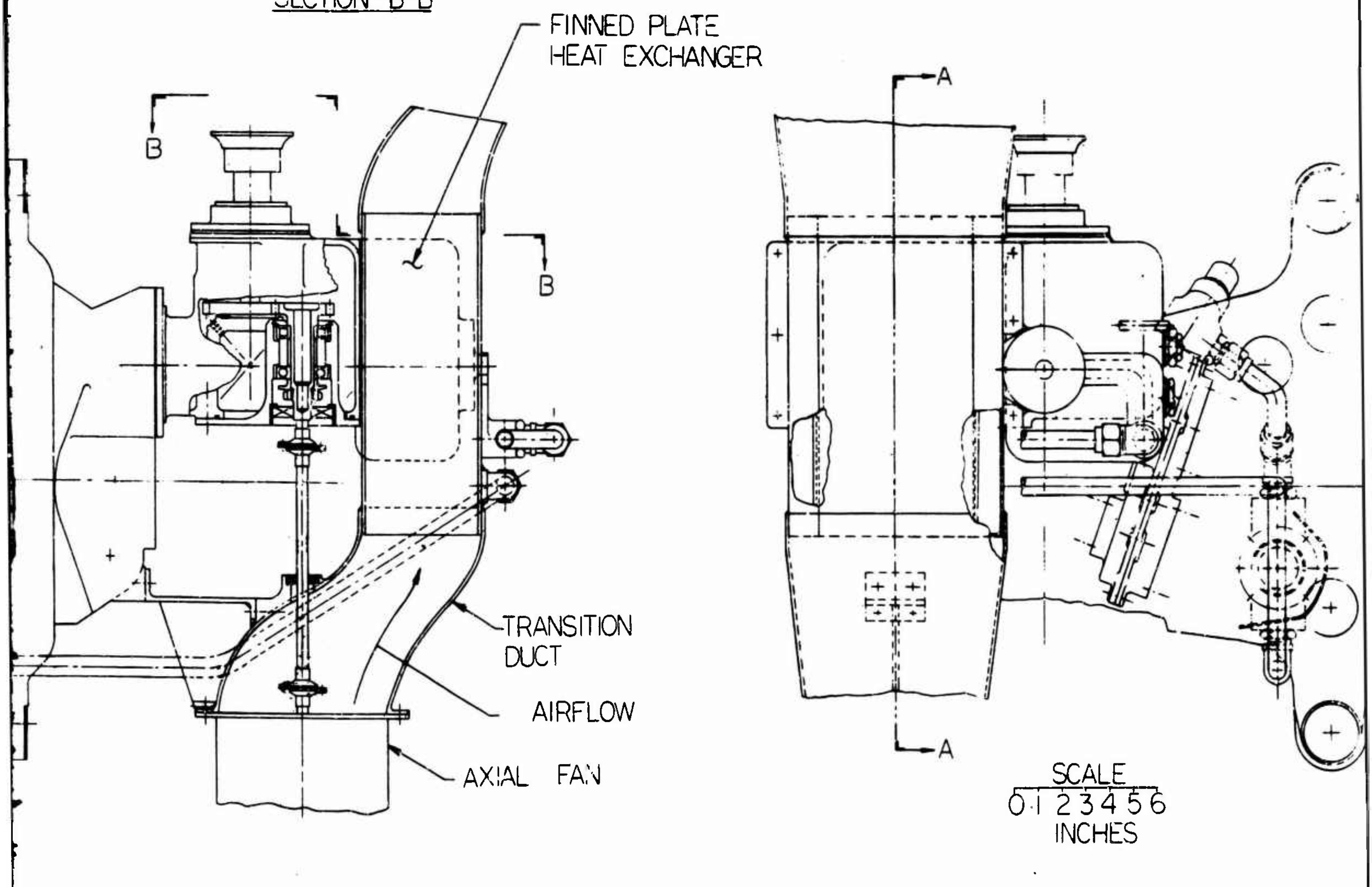


SECTION A-A

Figure 25. UH-1D Oil Sump Cooling System.


C
OIL FROM HEAT EXCHANGER
OIL TO HEAT EXCHANGER
C

SECTION B-B


FINNED PLATE HEAT EXCHANGER
B
B
A
TRANSITION DUCT
AIRFLOW
AXIAL FAN
A
SCALE
0 1 2 3 4 5 6
INCHES

B

TABLE XXVII. PERFORMANCE PARAMETERS - UH-1D OIL SUMP COOLING SYSTEM

Parameter	Value
Oil Temperature Into Heat Exchanger (°F)	280
Oil Temperature Into Gearbox (°F)	267.5
Ambient Air Temperature, Maximum (°F)	104
Adiabatic Heat of Compression Across Blower (°F)	6
Actual Air Temperature Into Heat Exchanger (°F)	110
Heat Exchanger Oil Heat Load (Btu/min)	590
Air Mass Flow Rate (lb/hr)	1,632
Oil Mass Flow Rate (gpm)	12.5
Heat Exchanger Effectiveness	.6
Air-Side Heat Transfer Area (ft^2)	29.8
Oil-Side Heat Transfer Area (ft^2)	10
Cross-Sectional Air-Side Area (ft^2)	.14
Air Mass Velocity (lb/hr ft^2)	12,000
Air-Side Reynolds Number	4,200
Air-Side Film Heat Transfer Coefficient (Btu/hr ft^2 °F)	14.2
Air Velocity Through Core (ft/sec)	75
Oil Mass Velocity (lb/hr ft^2)	14,000
Oil-Side Reynolds Number	490
Oil-Side Film Heat Transfer Coefficient (Btu/hr ft^2 °F)	200
Overall Heat Transfer Coefficient (Btu/hr ft^2 °F)	13.2
Calculated Heat Exchanger Core Length (ft)	1.0
Total Air-Side Pressure Head (in. of H_2O)	10.0
Blower Air Horsepower (HP)	1.0
Blower Shaft Horsepower (HP)	1.5
Oil-Side Pressure Drop (psi)	1.0

TABLE XXVIII. WEIGHT SUMMARY - UH-1D OIL SUMP COOLING SYSTEM

Cooling System Component	Weight (lb)
Heat Exchanger, Aluminum	10.2
Heat Exchanger Support Structure	3.1
Air-Side Ducting	4.9
Oil-Side Plumbing and Accessories	3.9
Blower	15.0
Blower Drive and Shafting	11.2
Total System Dry Weight	48.3

CONCLUSIONS

1. The cooling systems for the lubricating oil of helicopter transmissions, including those as large as the CH-54A main gearbox, can be made integral with the gearbox, thus greatly reducing that component's vulnerability to small-arms fire.

2. The integration of the transmission cooling system can be accomplished by a relatively modest redesign on several helicopters in the U. S. Army inventory that currently use remotely located external oil coolers. It can best be accomplished, however, by its consideration/requirement in the initial design of the helicopter transmission system.

3. On the basis of this investigation, which evaluated medium and large U. S. Army helicopters, an integral transmission cooling system, consisting of an oil-to-air heat exchanger located within the rotor shaft of the transmission, provides the maximum degree of invulnerability. The incorporation of oil distribution lines cast integrally (cored) within the main housing(s) (as incorporated in the CH-53A) further reduces the oil distribution system vulnerability.

4. The shell and tube configuration is the most efficient heat exchanger design for a rotor shaft cooler. This arrangement results in the minimum air-side pressure losses and power requirements for the air blower (fan). On new designs, the rotor shaft and cooler can be "sized" to provide the heat exchanger with an air-side pressure drop compatible with current remote external cooler designs, thus reducing blower requirements to external cooler system levels.

5. On those helicopters where the transmission design (i.e., size) does not permit the location of the gearbox cooler within the rotor shaft, a relatively high degree of invulnerability to small-arms fire can be achieved by integrating the transmission cooler within the oil sump. In this design, the heat exchanger should be mounted to, but capable of being separated from, the oil sump housing for maximum maintainability.

6. While designs for both rotor shaft and oil sump coolers are presented for the CH-54A, UH-1D, and CH-47A, this study indicates that the rotor shaft cooling system is the most effective system for the main gearboxes of the first two aircraft. An integral cooling system incorporating an oil sump cooler is recommended for the forward and aft transmissions of the CH-47A helicopter.

7. Transmission cooling provided by precooling inlet air to the heat exchanger or vapor cycle refrigeration has been found, on the basis of this investigation, to be considerably less efficient, reliable, and maintainable than simple, integral air-oil systems. While the vapor cycle system may, in some instances, be less vulnerable than the externally located air-oil systems currently used, the inclusion of an intermediate heat exchanging agent significantly decreases the efficiency and maintainability of the system.

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APPENDIX I

ANALYSIS - ROTOR SHAFT COOLING SYSTEM

INTRODUCTION

The analysis of any heat exchanging system is necessarily iterative. Initial calculations are performed for various air-side parameters, and the results are traded off until performance and air-side pressure drop can be optimized. The analysis contained herein is the final optimum result of the iterative procedure. Explanatory discussion is included in the calculation where necessary for reference. The analysis is for the shell and tube bundle core configuration located within the main rotor shaft, but the procedure is typical for any core configuration, including the finned-plate heat exchanger mounted integrally with the oil sump.

DESIGN ANALYSIS

The controlling parameter of this design is gearbox operating temperature or oil-out temperature. This is established at 280°F for this integral cooling design for the reasons enumerated in the Preliminary Design section of this report. The maximum oil mass flow rate is taken at 30 gpm (214 lb/min).

There are only four parameters that are absolutely fixed for the CH-54A that are considered to be nonvariables. These are presented in Table XXIX. All other parameters are calculated.

TABLE XXIX. INITIAL CONDITIONS - CH-54A COOLING SYSTEM

Parameter	Value
Oil Temperature Into Cooler (°F)	280
Outside Ambient Air Temperature (°F)	104
Oil Mass Flow Rate, Maximum (lb/min)	214
Oil Heat Rejection Rate (Btu/min)	3,905

The log-mean-temperature-difference (LMTD) method of analyzing heat exchangers is useful only when all the inlet and outlet temperatures can be

calculated or experimentally determined. In this design, only inlet oil temperature and inlet air temperature are known quantities, and a programmed trial-and-error analysis would have to be performed because of the logarithmic function involved. For these reasons, the effectiveness method is employed.

This method is based on experimental and prototype effectiveness data gathered and graphically presented for similar detail heat exchanger configurations on the ability of a certain configuration to transfer a given amount of heat.

When determining the maximum possible heat transfer for a perfect exchanger, the maximum value is obtained, by definition, when one of the fluids undergoes a temperature change equal to the maximum temperature differential inherent in the exchanger; i.e., the differential between the inlet temperatures of the hot and cool fluids. Since the energy balance between the fluids ensures that the total energy rejected by one fluid is received by the other fluid, the fluid having the smallest product of specific heat and mass flow rate must undergo the maximum temperature differential, or

$$\dot{Q}_{max} = \dot{m}_a c_p \Delta T_{max} \tag{20}$$

If the exchanger could be made to be 100 percent effective, the maximum possible air-side temperature differential T_{max} would equal the hot oil in temperature less the cool air in temperature, or

$$\Delta T_{max} = (280 - 104) \tag{21}$$

$$= 176^{o}$$

The effectiveness of a particular core configuration is determined from the following relations:

$$e = \frac{\dot{Q}}{\dot{Q}_{max}} \tag{22}$$

$$\dot{Q} = \dot{m}_a c_p \Delta T \tag{23}$$

$$e = \frac{\Delta T}{\Delta T_{max}} \tag{24}$$

From Reference 4, a graphical presentation of effectiveness versus number of transfer units (NTU) for a counterflow heat exchanger of similar configuration is reproduced in part in Figure 26.

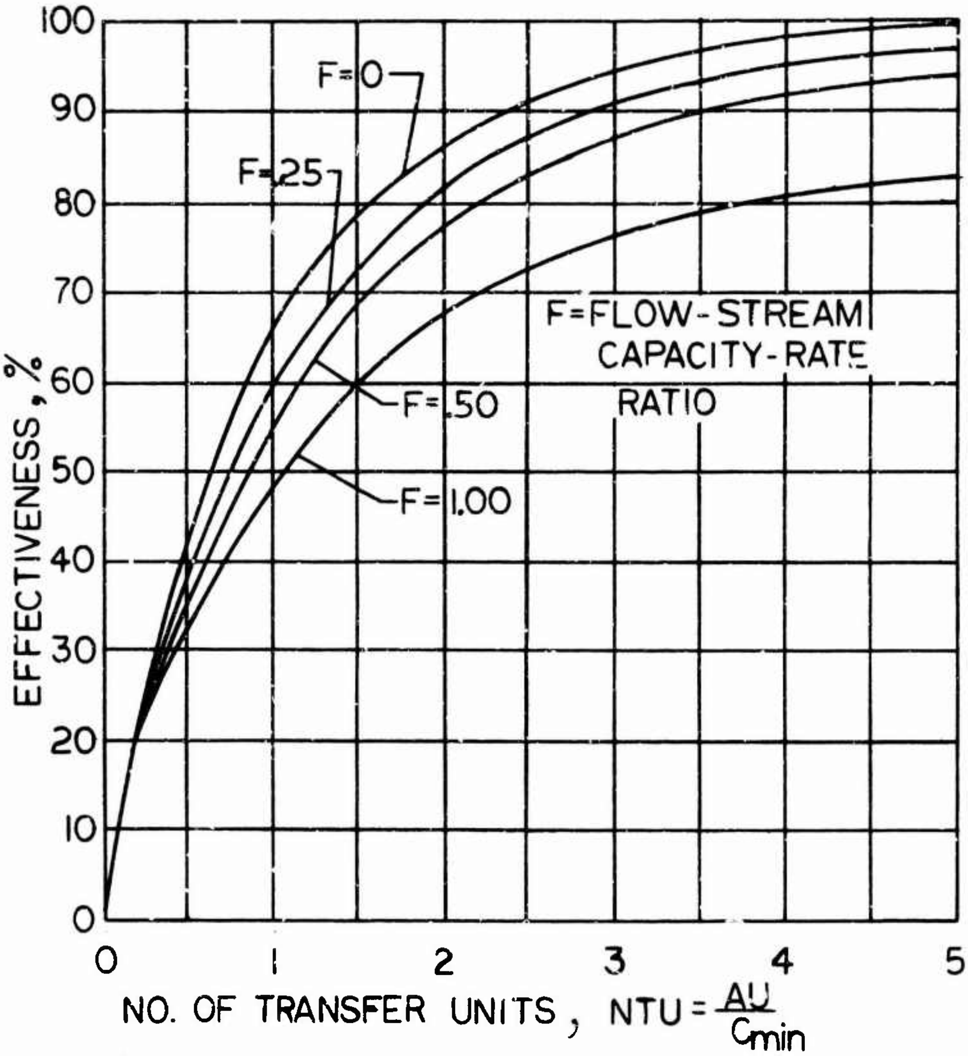


Figure 26. Effectiveness Versus Number of Transfer Units - Counterflow Exchanger.

For the integral heat exchanger located within the main shaft, air-side cross-sectional area is limited to the available bore of the main shaft diminished by exchanger structure, oil-side area, and exchanger-to-shaft clearance requirements. The air-side design parameters are critical. The small area gives rise to high air velocities and high blower horsepower, which decrease the attractiveness of the concept. Therefore, in trading off the parameters that determine heat exchanger effectiveness, in an effort to reduce air-side horsepower at the expense of physical size and weight of the exchanger itself, a high effectiveness requirement, such as .80, is imposed on the exchanger design.

Since air-side pressure drop varies with the square of mass flow rate and directly with the length (or heat transfer area) of the exchanger, a long cooler with a larger number of transfer units and high effectiveness is more desirable than a smaller cooler with a fewer number of transfer units and lower effectiveness. In trading off the entire main mast cooling system including blower and blower drive requirements, the longer, more effective cooler is more desirable. Solving equation (24) for ΔT_{act} yields the following relation:

$$\Delta T_{act} = e\,\Delta T_{max}$$

$$= (.80)\ (176)$$

$$= 141^{\circ}F$$

$$\dot{m}_a = \frac{\dot{Q}}{c_p\ \Delta T_{act}}$$

$$= \frac{3905}{(.241)\ (141)}$$

$$= 115\ \frac{lb}{min}$$

The dimensionless ratio C_{min}/C_{max} is introduced here and is defined as the flow-stream capacity-rate ratio, F.

$$C_{min} = \dot{m}_a\ c_p \tag{25}$$

$$= (115)\ (60)\ (.241)$$

$$= 1660\ \frac{Btu}{hr\ ^{o}F}$$

$$C_{max} = \dot{m}_o\ c_p \tag{26}$$

$$= (214)\ (60)\ (.53)$$

$$= 6820\ \frac{Btu}{hr\ ^{o}F}$$

$$F = \frac{C_{min}}{C_{max}} \tag{27}$$

$$= \frac{1660}{6820}$$

$$= .244$$

$$\cong .25$$

From Figure 26, the NTU value required equals 1.8. By definition, the following relation is written:

$$UA_{reqd} = (NTU)\,(C_{min}) \tag{28}$$

$$= (1.8)\,(1660)$$

$$= 3000 \; \frac{Btu}{hr\ ^oF}$$

This represents the index of the exchanger performance required of the counterflow heat exchanger operating under the given air-side and oil-side conditions, with an effectiveness of .80. Once UA_{reqd} has been determined for the configuration, the geometry of the heat exchanger and its effect on the heat transfer area is calculated.

The shell and tube bundle cross-sectional configuration is depicted in Figure 10. The tubes are spaced in an equilateral pattern, .150 inch between centers (.150-inch pitch). The stainless steel tubes have a .125-inch outside diameter and a .005-inch wall thickness. The maximum number of tubes that can be installed within the shell inside diameter of 6.815 inches on the equilateral pattern is 1740. This number is diminished by the oil inlet and outlet lines to the manifolds of the heat exchanger, which occupy approximately 60 air tube spaces. The cross-sectional open air-side area of all air tubes is calculated as follows:

$$A_{cs} = \frac{\pi(.115)^2\,(1680)}{4}$$

$$= 17.4 \text{ in.}^2$$

The heat transfer area of all tubes on the air-side per unit length of heat exchanger is calculated as follows:

$$A_a = \frac{\pi(.115)\,(1680)}{12}$$

$$= 50.5 \frac{ft^2}{ft}$$

The calculation of the air-side and oil-side film heat transfer coefficients is performed as follows:

$$G_a = \frac{\dot{m}_a}{A_{cs}} \tag{29}$$

$$= \frac{(115)(60)(144)}{17.4}$$

$$= 57,200 \frac{lb}{hr\ ft^2}$$

$$Re_a = \frac{G_a D_h}{\mu} \tag{30}$$

$$= \frac{(57,200)(.115)}{(12)\quad(.052)}$$

$$= 10,500 \text{ turbulent}$$

By definition, the generalized heat transfer grouping, j_a, when plotted against the Reynolds number, defines the inherent heat transfer characteristics of the heat exchanger surfaces. The value of j_a is obtained from Reference 4.

$$j_a = (St)(Pr)^{2/3} \tag{31}$$

$$= \left[\frac{\bar{h}_a}{G_a c_p}\right]\left[\frac{\mu c_p}{k}\right]^{2/3}$$

When the calculated Reynolds number is introduced and a ratio L/D_i from the iterative procedure of approximately 117 is introduced, j_a is determined to be .0031. Rewriting equation (31) yields the following expression:

$$\bar{h}_a = j_a G_a c_p (Pr)^{-2/3}$$

$$= (.0031)(57,200)(.241)\left[\frac{(.052)(.241)}{.018}\right]^{-2/3}$$

$$= 54.2 \frac{Btu}{hr\ ft^2\ {}^oF}$$

The mass rate of flow of oil through the current production CH-54A main transmission oil cooling system is 17.5 gpm. For the integral cooling systems investigated during this study, the maximum available mass flow rate of 30 gpm from the pump is circulated through the heat exchanger. From the calculation of $\bar{h}_o$ that follows, it becomes apparent that a high G_o value will yield a high h_o value, and a high $\bar{h}_o$ will help to increase the overall heat transfer coefficient. When the air-side $\bar{h}_a$ is low, every effort is made to get the other $\bar{h}$ terms up so that U can approach $\bar{h}_a$, as is the case in this analysis. From the effectiveness curves presented in Figure 26, the number of transfer units is optimized by making the ratio C_{min}/C_{max} small, for a certain heat exchanger effectiveness; or, when C_{min}/C_{max} is small, heat transfer capability is maximized. It is more effective to cool a large amount of oil through a smaller temperature differential than it is to cool a smaller amount of oil through a large temperature differential.

There are no particular disadvantages to flowing 30 gpm of oil through this system, and there is the distinct advantage of improved exchanger performance.

The current production CH-54A lubrication system components can be used with no major modifications required. The main lube pump has a 30-gpm capability now at current pump rpm. 12.5 gpm is presently bypassed through a bypass circuit to the inlet side of the pump. 17.5 gpm is circulated through the radiator to the distribution system. In the proposed system, simply by relocating the oil bypass valve assembly from the upstream to the downstream side of the heat exchanger, the cooled 12.5 gpm of bypass oil is dumped into the sump, where it mixes with hot oil return from the gearbox. The cooled 17.5 gpm is routed to the distribution system in the same manner as on the current production system.

The shell-side segmented baffling causes the oil to circulate back and forth over the air tubes as the oil progresses up the heat exchanger. The minimum oil-side free-flow area will occur at the leading edge of the segmented baffle where the oil changes direction transversely, and it will equal the spacing between air tube outside walls times the number of tubes in the tube row at the baffle leading edge times the height or baffle spacing.

$$
\begin{aligned}
A_{os} &= (St)\,(Nb)\,(bt) \qquad (32)\\
&= (.025)\,(35)\,(2.0)\\
&= 1.75\ \text{in.}^2
\end{aligned}
$$

$$G_o = \frac{\dot{m}_o}{A_{os}} \tag{33}$$

$$= \frac{(214)\ (60)\ (144)}{1.75}$$

$$= 1,060,000 \ \frac{\text{lb}}{\text{hr ft}^2}$$

The external (oil-side) hydraulic diameter, D_h, for fluid flow normal (transverse) to the axis of a bank of packed tubes for turbulent flow approximates D_o from Reference 4.

$$R_{e_o} = \frac{G_o\ D_h}{\mu} \tag{34}$$

$$= \frac{(1,060,000)\ (1.25)}{(12)\ (9.85)}$$

$$= 1121$$

From Reference 5, the oil-side film heat transfer coefficient is calculated from the following empirical relation for similar flow conditions:

$$\bar{h}_o = \frac{0.22\ c_p\ G_o}{(R_{e_o})^{.4}\ (P_r)^{2/3}} \tag{35}$$

$$= \frac{(0.22)\ (.53)\ (1,060,000)}{(1121)^{.4} \left[\frac{(.53)\ (9.85)}{.08}\right]^{2/3}}$$

$$= 460 \ \frac{\text{Btu}}{\text{hr ft}^2\ {}^oF}$$

The overall heat transfer coefficient is calculated as a reciprocal summation of the individual film coefficients and transfer coefficients of the elements involved in transferring heat from the oil to the air. There are no fouling factors or dirt factors involved in this summation for the following reasons:

The airflow through the tubes is of such a high mass velocity that air-side fouling, even with dirty air, is not anticipated. The tubes are clean and smooth and have no internal finning or dimpling. The air will have a self-cleaning effect on the tubes; in fact, with dirty air, erosion, not fouling,

will possibly be the problem. Therefore, no fouling or dirt factor on the air side is calculated.

The oil flowed through the core is passed through a multielement filter unit of 40-micron porosity. Only absolutely clean oil is passed through the exchanger. All sludge, wear particles, and carbonaceous compounds present in the gearbox oil are filtered out. In addition, the rotating components of the gearbox that have internally fed lubrication provisions, such as camshafts, transfer tubes, and hollow gear shafts, act as oil centrifuges, and sludge generated by wear is plated out on these parts before it enters the lubrication filtering system. The mass velocity of the oil side is very high, and the heat exchanger oil flow passage outside the tubes is smooth, clean, and free from external finning. Therefore, no fouling or dirt factor on the oil side is calculated.

The investigator's experience, in conjunction with the radiator manufacturer's experience, indicates that all transmission system oil coolers do not develop a dirt layer internally in service unless the filter bypasses because of contamination. Oil coolers subjected to contamination from a bypassing filter are rejected for further use.

The general expression for the overall heat transfer coefficient is written as follows:

$$\frac{1}{U} = \frac{1}{\bar{h}_o} + \frac{1}{\bar{h}_{ofoul}} + \frac{x_w}{k} + \frac{1}{\bar{h}_a} + \frac{1}{\bar{h}_{afoul}} \tag{36}$$

Eliminating fouling and dividing through by the appropriate area terms yields the following expressions:

$$\frac{1}{UA/ft} = \frac{1}{\bar{h}_o A_o} + \frac{x_w}{K A_w} + \frac{1}{\bar{h}_a A_a} \tag{37}$$

where $A_o = \pi D_o N$

$$= \frac{\pi(.125)(1680)}{12}$$

$$= 55 \frac{ft^2}{ft}$$

$$\frac{1}{(UA/ft)} = \frac{1}{(460)(55)} + \frac{.005}{(12)(9.4)(52.7)} + \frac{1}{(54.2)(50.5)}$$

$$UA/ft = 2470 \frac{Btu}{hr\ ft\ ^oF}$$

The required length of the heat exchanger <u>core</u> is calculated from the following relation:

$$L = \frac{UA_{reqd}}{UA/ft} \tag{38}$$

$$= \frac{3000}{2470}$$

$$= 1.22\ ft$$

As initially analyzed, the integral cooling system located within the main shaft had an airflow entering the blower, flowing upward through the heat exchanger, and exiting to atmosphere at the top of the main shaft. Because of the high pressure drop experienced across the heat exchanger, the blower horsepower requirement is relatively large, and a significant amount of heat was imparted to the airstream by the blower from work done on the air, combating friction, turbulence, and compression. As a result, the inlet temperature to the heat exchanger was equal to the sum of ambient temperature plus this temperature rise across the blower. For the configuration utilizing a pusher-type blower, the system was marginal. In the draw-through system, as proposed in this report, ambient air is drawn in from the top of the main shaft and passes through the heat exchanger before passing through the blower. While the blower has to pass more cubic feet of air per unit time out of the exchanger because of the lowered air density at elevated temperature, the system becomes fully feasible heat-transferwise.

The air-side pressure drops across the draw-through system are calculated in the order in which they occur, as follows:

The pressure drop across the inlet converging nozzle is expressed as follows:

$$\Delta P_{nozzle} = \rho_1 \frac{(V_2^2 - V_1^2)}{2g} \tag{39}$$

where $$\rho_1 = \frac{P}{R\ T} \tag{40}$$

and $$V = \frac{\dot{m}_a}{\rho A_{cs}} \tag{41}$$

$$\Delta P_{nozzle} = \frac{(.07)\ (108)^2}{(2)\ (32.2)}$$

$$= 12.7 \frac{lb}{ft^2}$$

$$= 2.5 \text{ in. of } H_2O$$

The pressure drop equation across the heat exchanger, including the abrupt changes at the end faces, is expressed as follows (each group of terms within the equation has physical significance and is treated separately):

$$\Delta P_{core} = \frac{G_a{}^2}{2g\ \rho_1} \text{ (sudden contraction term + flow acceleration term + core friction term - sudden expansion term)}$$

$$= \frac{G_a{}^2}{2g\ \rho_1} \left(K_c + 1 - \omega^2\right) + 2\left(\frac{\rho_1}{\rho_2} - 1\right)$$

$$+ \frac{f_d\ A_a\ L\rho_m}{A_{cs}\ \rho_2} - \left(1 - \omega^2 - K_e\right) \frac{\rho_1}{\rho_2} \qquad (42)$$

where K_c, K_e, and f are obtained from Reference 4.

$$\Delta P_{core} = \left[\frac{(57,200)^2}{(2)\ (4.17 \times 10^8)\ (.07)}\right] \left[\left[.22 + 1 - \left(\frac{17.4}{36.4}\right)^2\right]\right.$$

$$+ 2\left[\frac{.07}{.048} - 1\right] + \left[\frac{(.0072)\ (50.5)\ (1.22)\ (144)\ (.059)}{(17.4)\ (.048)}\right]$$

$$\left.\left(1 - \left[\frac{17.4}{36.4}\right]^2 - .22\right)\left[\frac{.07}{.048}\right]\right]$$

$$\Delta P_{core} = 313 \frac{lb}{ft^2}$$

$$= 60.3 \text{ in. of } H_2O$$

The pressure drop through the transition duct is made up of three components as shown:

$$\Delta P_{duct} = \text{(friction term + curvature terms - diffuser term)}$$

The airflow through the transition duct is of such a low Mach number and static pressure that the flow is considered to be incompressible. The

curvature terms are made up of the losses from the S-shaped reverse curve transition duct, which are approximated by two 45° elbows of the appropriate radii of curvature and mean diameters.

$$\Delta P_{duct} = \frac{\rho_1}{2g}\left[\frac{f\,L_e\,V_m^2}{D_{i_m}} + K_{e_1}\,V_{m_1}^2 + K_{e_1}\,V_{m_2}^2 - \eta\left(V_1^2 - V_2^2\right)\right] \tag{43}$$

where f and K_{e_1} are obtained from Reference 4, and η is obtained from Reference 6.

$$\Delta P_{duct} = \frac{.048}{(27)(32\ 2)}\left[\frac{(.014)\,(18)\,(145)^2}{7.84} + (0.42)\,(110)^2 + (0.42)\,(178)^2 - (.80)\,(212^2 - 77^2)\right]$$

$$= -9.1\ \frac{\text{lb}}{\text{ft}^2}$$

$= -1.8$ in. of H_2O pressure rise

The flow through the diffuser acts like a pump to raise flow-stream pressure at the expense of kinetic energy, and it is subtracted from head felt by the blower.

The pressure drop due to velocity head is calculated next. For a fluid flowing with a certain velocity, namely, the discharge velocity of the blower, the pressure created by velocity impingement on stator and rotor vanes is called the velocity pressure, or velocity head, and is the second component of total pressure head in fan law engineering. The velocity head is expressed analytically as follows:

$$\Delta P_{velocity} = \frac{\rho\,V_2^2}{335} \tag{44}$$

$$= \frac{(.048)\,(77)^2}{335}$$

$= .85$ in. of H_2O

Total air-side pressure drop across the system is the algebraic summation of the individual drops.

$$\Delta P_{total} = 2.5 + 60.3 - 1.8 + .85$$

$$= 61.9 \text{ in. of } H_2O$$

Two other minor flowstream interruptions in the systems which contribute a slight resistance to airflow were evaluated. The portion of the inlet and outlet oil lines from the exchanger crossing the flowstream, and the blower drive shaft entering the transition duct and crossing the flowstream were considered. The resistance was calculated by analyzing airflow at right angles to a slender cylinder in the flowstream. The resistances are negligible.

Also, on the discharge side of the blower where the hot air is dumped to atmosphere, a diffuser analysis was conducted to evaluate reduction of blower air horsepower. This is accomplished by doing some of the work of bringing the system pressure (decreased by 61.9 inches of water going into the blower) up to atmospheric pressure at discharge, in a long diffuser. The analysis indicates that the optimum diffuser, which would approach 19 inches in length and which would add considerable overhanging mass to the blower and support structure, could decrease shaft power at best by only approximately 3 HP. For these reasons, no diffuser exit section is provided as part of the preliminary design.

The expression for air horsepower in axial vane-type blowers is written as follows:

$$AHP = \frac{(62.3\ \Delta P_{total}\ m_a)}{(12)\ (33,000)\ (\rho_1)} \tag{45}$$

$$= \frac{(62.3)\ (61.9)\ (115)}{(12)\ (33,000)\ (.048)}$$

$$= 23.4 \text{ HP}$$

In the calculation of shaft power requirements to the blower, from Reference 7 and vendor consultation, the blower efficiency is taken conservatively at 70 percent. Shaft horsepower is then calculated from the following relation:

$$SHP = \frac{AHP}{\eta} \tag{46}$$

$$= \frac{23.4}{.7}$$

$$= 33.4 \text{ HP}$$

The oil-side pressure drop across the shell and tube-type heat exchanger per baffle pass is calculated from the following relation:

$$\Delta P_o = \frac{2 f_o B_o N_b G_o^2}{g\rho} \tag{47}$$

where $\quad f_o = a_o (Re)^{-.15}$ (48)

$$a_o = 0.23 + \frac{0.11}{(x_t - 1)^{1.08}}$$

$$x_t = \frac{.150}{.125}$$

$$a_o = 0.23 + \frac{0.11}{(1.20-1)^{1.08}}$$

$$= .855$$

$$f_o = (.855)\ (1121)^{-.15}$$

$$= .297$$

$$\Delta P_o = \frac{(2)\ (.297)\ (1.1)\ (20)\ (1,060,000)}{(4.17 \times 10^8)\ (53.5)\ (144)}$$

$$= 4.57 \text{ psi per baffle pass}$$

There are six segmented baffles and seven baffle passes on the 2-inch baffle spacing pattern over the core length of 1.22 feet. Total oil-side pressure drop is calculated as follows:

$$\Delta P_{o_{total}} = (4.57)\ (7)$$

$$= 32 \text{ psi}$$

This is the total oil-side pressure head that must be overcome by the main lubrication pump. The pressure drop in the current production CH-54A oil cooler radiator is 15 psi. The current production lubrication pump and plumbing system can easily accommodate a core pressure drop of 32 psi with no modification or redesign. Actual pump output pressure is simply

increased to provide the same required final oil pressure at the oil jets downstream from the integral oil cooler.

The stainless steel full hard air tubes are subjected to crushing forces from the oil on the shell side of the tube bundle. While actual oil-side working pressure for the CH-54A integral oil cooling system will be approximately 80 psi at the inlet, in extreme cold weather operation, due to the high oil viscosity on start-up, cooler oil pressure may approach 200 psi initially before stabilizing. A crushing proof-pressure calculation is performed to certify the structural integrity of the tubes on the shell side for cold start-up conditions. For round tubes with large L/D_0 ratios, the expression for critical pressure is written as follows:

$$\Delta P_c = \frac{E}{2(1 - \nu^2)} \left[\frac{x}{D_m} \right]^3 \tag{49}$$

$$= \frac{(26 \times 10^6)}{2(1 - .10^2)} \left[\frac{(.005)}{.120} \right]^3$$

$$= > 10,000 \text{ psi}$$

APPENDIX II

ANALYSIS - VAPOR CYCLE REFRIGERATION SYSTEM

INTRODUCTION

The analysis of the vapor compression refrigeration cycle is basically a trade-off study of the performance of the subcomponents of the system operating at different state points (temperature, pressure, enthalpy, etc.). Performance analysis using different refrigerants to obtain different operating characteristics is also made in an effort to minimize the size of the condenser. The analysis contained herein is the final iteration covering the vapor cycle system using Refrigerant 12, as described in the Preliminary Design section. The analysis is typical for all vapor compression cycles using any refrigerant.

DESIGN ANALYSIS

The vapor cycle refrigeration system is analyzed by establishing the state points of each of the flow processes of the thermodynamic path shown in Figure 27.

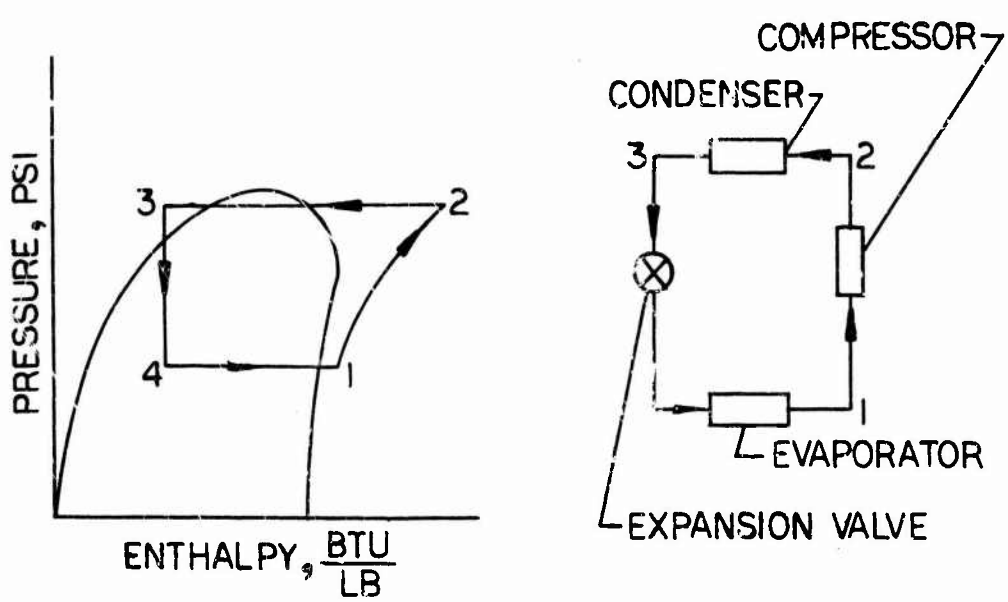


Figure 27. Vapor Cycle Pressure - Enthalpy Diagram.

From an iterative analysis, an evaporator temperature of 160°F yields an optimum trade-off of system compressor shaft work and evaporator heat transfer performance and physical size for Refrigerant 12. The heat transfer analysis of oil to refrigerant in the tube bundle evaporator is similar to that performed for the rotor shaft oil-to-air heat exchanger in Appendix I and therefore is not repeated.

From the thermodynamic properties of saturated Refrigerant 12 at 160°F in the evaporator, assuming evaporation at constant pressure, $P_4 = P_1 =$ 279.8 psi. In all calculations, 10°F of superheat and 10°F of subcooling across the condenser are taken, based on experience. Then T_1 = 160°F + 10°F superheat = 170°F. Using the tables for superheated vapor at T_1 = 170°F and P_1 = 279.8 psi, h_1 = 93.34 Btu/lb and s_1 = 0.1643 Btu/lb°R.

For isentropic compression from state points 1 to 2 in the compressor, $s_2 = s_1$ = 0.1643 Btu/lb°F.

In the vapor cycle system, the controlling parameters are maximum condenser and evaporator temperatures. Because the critical temperature for Refrigerant 12 is 233.6°F, the maximum condenser temperature is established at 220°F. For a condensing temperature of 220°F, condenser pressure for saturated liquid from the tables, P_2 = 524.4 psi. Assuming condensing at constant pressure, $P_3 = P_2$ = 524.4 psi. Interpolating the tables at state point 2, h_2 = 97.79 Btu/lb.

At state point 3, P_3 = 524.4 psi, the tables for 10°F subcooled liquid in the condenser at T_3 = 220°F - 10°F = 210°F and h_3 = 62.96 Btu/lb. Free expansion from point 3 to point 4 in the throttle occurs ideally at constant enthalpy. Then $h_4 = h_3$ = 62.96 Btu/lb.

The change in enthalpy across the evaporator is expressed as follows:

$$\Delta h_{evap} = h_1 - h_4 \tag{50}$$

$$= 93.34 - 62.96$$

$$= 30.38 \frac{\text{Btu}}{\text{lb}}$$

In the evaporator, the oil heat load of 3905 Btu/min must be absorbed.

$$\dot{m}_r = \frac{\dot{Q}}{\Delta h_{evap}} \tag{51}$$

$$= \frac{3905}{30.38}$$

$$= 128.5 \ \frac{lb}{min}$$

The increase in enthalpy in the compressor is calculated from the following relation:

$$\Delta h_{comp} = h_2 - h_1 \qquad (52)$$

$$= 97.79 - 93.34$$

$$= 4.45 \ \frac{Btu}{lb}$$

The theoretical horsepower required to drive the compressor is calculated as follows:

$$HP_{comp} = (\Delta h)\ (\dot{m}_r) \qquad (53)$$

$$= \frac{(4.45)\ (128.5)}{42.44}$$

$$= 13.5 \text{ HP}$$

Compressor efficiency is determined from curves published by helical screw compressor manufacturers and vendor consultation as 54 percent. The actual compressor horsepower is then calculated as follows:

$$SHP_{comp} = \frac{HP_{comp}}{\eta} \qquad (54)$$

$$= \frac{13.5}{.54}$$

$$= 25 \text{ HP}$$

The actual increase of enthalpy in the compressor is the sum of the effects from pressure rise and the compressor inefficiencies.

$$\Delta h_{comp_{act}} = \frac{h_2 - h_1}{\eta} \qquad (55)$$

$$= \frac{4.45}{.54}$$

$$= 8.25 \frac{\text{Btu}}{\text{lb}}$$

The actual enthalpy at point 2 will be 101.59 Btu/lb (93.34 + 8.25).

The heat load of the condenser is comprised of both the heat load of the evaporator and the heat of compression. It is calculated below.

$$\dot{Q}_{total} = (h_2 - h_3)\,\dot{m}_r \qquad (56)$$

$$= (101.59 - 62.96)\,(128.5)$$

$$= 4960 \frac{\text{Btu}}{\text{min}}$$

A complete heat transfer analysis of the condenser was performed but is not presented because the procedure is similar to the procedure detailed in Appendix I. For the condenser selected, the air pressure drop is calculated to be 33.2 inches of H_2O at an airflow of 224 lb/min, which is the minimum airflow practical for a 220°F condensing temperature.

The horsepower requirement for the condenser blower is calculated as follows:

$$SHP_{cond\ blower} = \frac{(62.3)\,(\Delta P_{cond})\,\dot{m}_r}{(12)\,(33,000)\,\rho\,\eta} \qquad (57)$$

$$= \frac{(62.3)\,(33.2)\,(224)}{(12)\,(33,000)\,(.055)\,(.7)}$$

$$= 30 \text{ HP}$$

Total refrigeration cycle power requirements are then equal to compressor requirements plus condenser blower requirements.

$$SHP_{total} = SHP_{comp} + SHP_{cond\ blower} \qquad (58)$$

$$= 25 + 30$$

$$= 55 \text{ HP}$$

APPENDIX III

OIL-FOG LUBRICATION/COOLING FOR HELICOPTER TRANSMISSIONS

FOREWORD

This appendix covers the investigation of an oil-fog lubrication and cooling system for helicopter transmissions. Pertinent empirical and analytical data upon which portions of this section are based are drawn from programs conducted by the following companies:

Esso Research and Engineering Company, Linden, New Jersey

Pratt and Whitney Aircraft Division of United Aircraft Corporation, East Hartford, Connecticut

INTRODUCTION

In an oil-fog lubrication system, oil is distributed to the rotating parts of the transmission system not as a liquid stream but as a very thin fog. Essentially, an oil fog (or micro-fog system) utilizes air as the carrier for the oil lubricant, which makes up only a few parts per million by volume of the air-oil stream. The oil droplets vary from about 0.25 to 1.7 microns in diameter. Since air is used as the primary coolant, the oil churning losses are greatly reduced. Basically, the most important requirement for an oil-fog lubrication system is a controlled flow of clean, dry air supplied in sufficient volume to meet the gearbox cooling requirements. This cooling effort must be accomplished over a wide range of ambient temperatures and at all practical operating altitudes.

SYSTEM DESCRIPTION

The helicopter transmission oil-fog lubrication system would consist of a compressed air source, an oil-fog generator, an oil supply tank, and the necessary ducting/lines to supply the air-oil fog to the rotating parts of the gearbox. In this system, the oil would be "expendable" (or used only once) and allowed to discharge to the atmosphere, dispersed in the fog, after providing its necessary lubrication function.

BACKGROUND

Recent studies of air-oil fog lubrication/cooling systems have been conducted by several investigators, including those listed in the foreword to this appendix. In their evaluation of the concept, Esso conducted tests on the Ryder gear tester. They report that in these tests:

"The gears were operated at speeds from 2,000 rpm to 10,000 rpm (peripheral speeds of 1800 ft/min to 9,200 ft/min) and at loads up to 200,000 psi Hertz. Fog air ranged up to 4.4 SCFM at 29 ppm oil, and cooling air ranged up to 68 SCFM. The scuff levels of the test gear teeth were as low as, and perhaps lower than, those obtainable with the same MIL-L-23699A oil in conventional liquid form -- 12.4 percent with fog as compared to 19.5 percent with liquid. Moreover, the gears were cool enough to touch immediately after stopping; whereas the gear case, containing conventionally lubricated bearings, was too hot to touch. The low gear temperature was due partly to the efficacy of the air cooling, but probably also owed something to reduced heat generation.

"This performance could probably be improved. The experiment was a preliminary one, and large safety factors were included. Clearly, much less cooling was actually needed, and we probably used more oil than necessary. Moreover, no attempt was made to optimize the lubricant composition. It is known that conventional oils suffer as much as 25 percent material loss when fogged, through formation of fog particles too small to wet out upon impingement -- so-called 'free-fogging.' This wastage can be reduced to approximately 2 percent by small amounts of Esso proprietary additives. Additionally, the usual oxidation inhibitors may not be needed at all, and a modified level of EP additives may be optimum for oil-fog lubrication. At these temperatures deposit formation should not be detectable in the normal lifetime of such gear units."

TRANSMISSION POWER LOSSES - OIL-FOG SYSTEM

To determine the feasibility of the use of an oil-fog lubrication/cooling system for high-power, high-performance helicopter power train components, an analytical evaluation was made for the CH-54A main gearbox requirements. The considerable amount of test data available for helicopter main transmissions from testing conducted on regenerative and dynamic system test facilities makes it possible to define power transmitted versus power loss (efficiency) curves for these gearboxes and to separate the components of mechanical inefficiency. Figure 28 shows these data for the CH-54A main gearbox. Of the total CH-54A gearbox losses of 5100 Btu/min, 1350 Btu/min is created by friction losses due to power transmitted, and the balance of 3750 Btu/min is due to windage and churning, as shown in Figure 28. For the purpose of this analysis, it has been assumed that approximately half of this total, or approximately 1900 Btu/min, is due to churning. The incorporation of an oil-fog system which eliminates churning will reduce the mechanical losses of the gearbox to 3200 Btu/min at hovering power, as indicated by the lower curve of Figure 28.

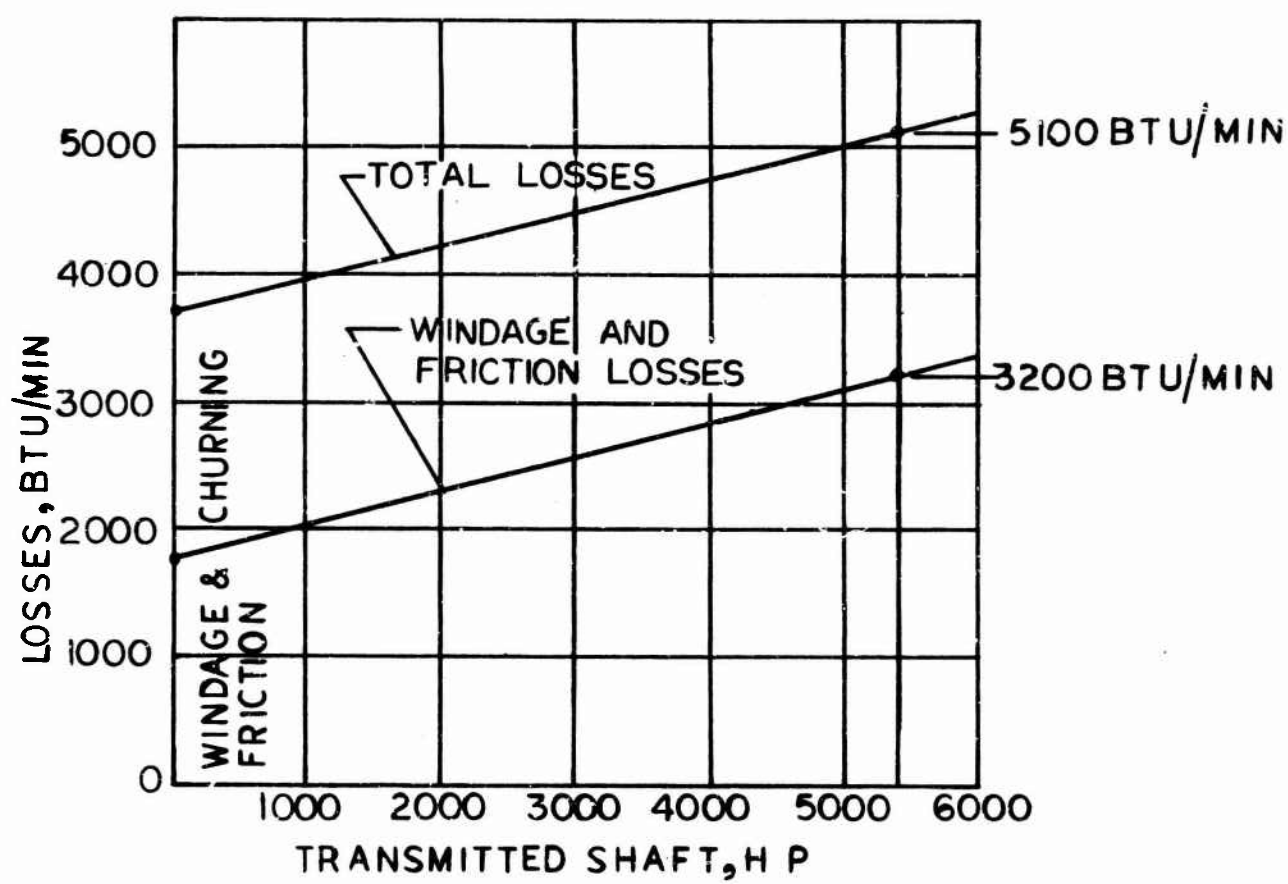


Figure 28. Main Gearbox Friction and Churning Losses (Based on Figure 6).

Cooling Air Mass Flow

From preliminary data obtained for oil-fog lubrication/cooling in tests conducted by Essc Research and Engineering on the Ryder gear test machine in which oil/air mist was used, a mean temperature rise for cooling air through the gearbox is approximately 40°F. This temperature rise is used in the following estimation of cooling airflow.

Referring to Figure 28, the losses due to friction and windage are 3200 Btu/min. Therefore, when T = 40°F and Q = 3200 Btu/min, the cooling airflow is calculated as follows:

$$\dot{m}_a = \frac{\dot{Q}}{C_p \Delta T} \tag{59}$$

$$= \frac{3200}{(.24)\ (40)} = 333 \text{ lb/min}$$

This quantity of cooling air will be supplied by a separately installed air compressor, which will be driven by the aircraft main gearbox or APU. While it might seem that a natural source of supply for the cooling air would be from the compressors of the helicopter's engines, this in fact is not feasible because the cooling air mass flow is too large a proportion of the engine total mass flow. To bleed this flow from the engine(s) would significantly reduce engine power output in critical flight conditions such as hover.

Power Requirements - Cooling Air Blower

It would be a very difficult, if not an impossible, task to calculate with any accuracy the pressure loss due to a mass of cooling air passing through the main gearbox of a helicopter transmission. This is due not only to the complex paths which the air takes (see Figure 29) but also to the additional effects of rotation and running clearance around the rotating components. If this research effort is continued, empirical data will be necessary to verify the assumptions made in estimating the power absorption of the cooling air blower.

For an initial estimate of pressure losses incurred, a simple analogy is taken. This approximates the resistance to airflow as an equivalent number of restrictions in series, the areas of which have been obtained from gearbox detail drawings.

The method used to estimate the overall pressure ratio (rotation of the dynamic components is not included) is somewhat analogous to a series of resistances (electrical, hydraulic, etc.). The gearbox therefore is put

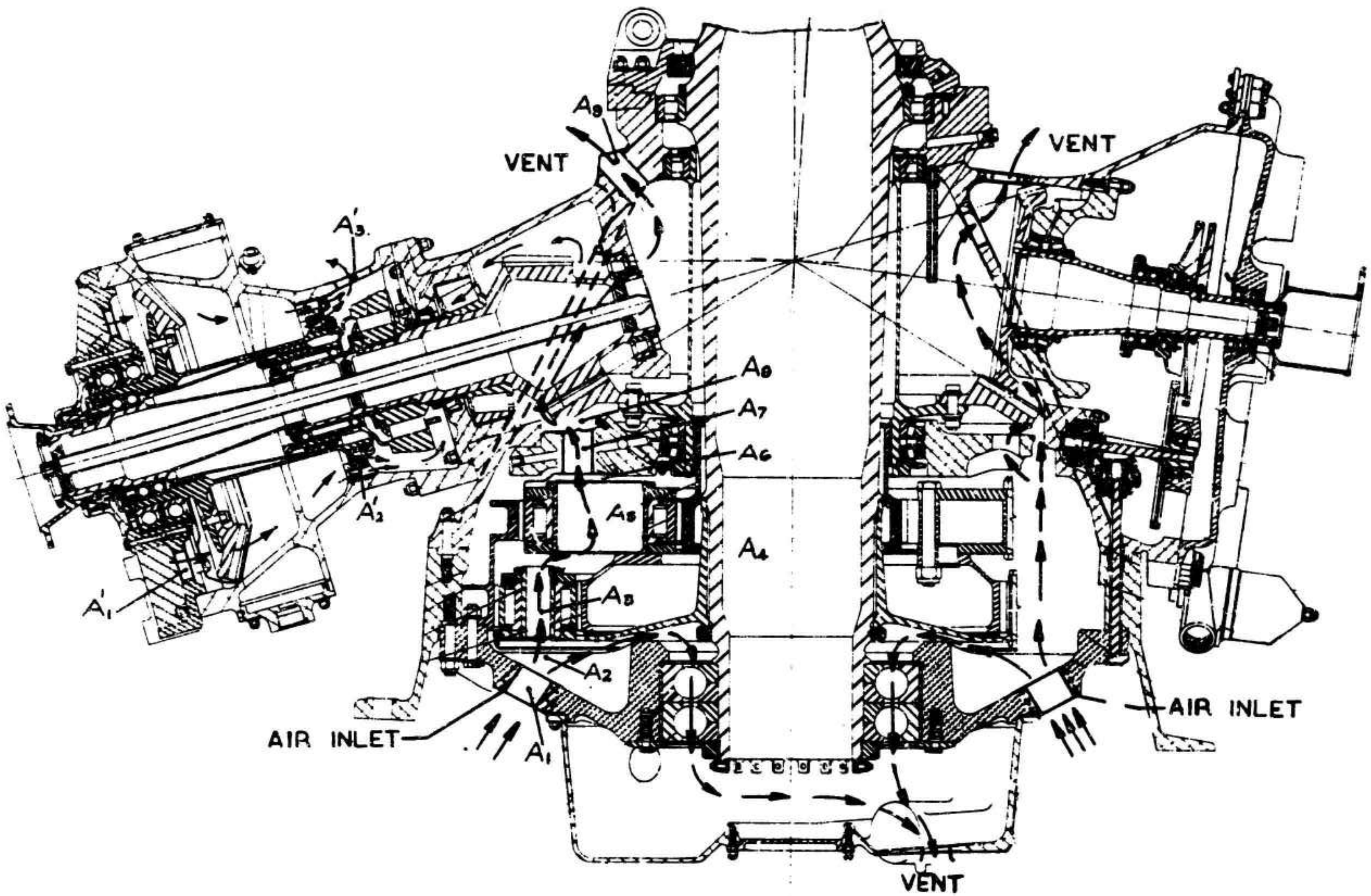


Figure 29. CH-54A Main Gearbox Cooling Airflow Path.

into the form of a series of restrictions with respective areas $A_1, A_2, \ldots A_n$, with pressures through the system of $P_1, P_2, \ldots P_{n+1}$,

$$P_1 \;|\; P_2 \;|\; P_3 \;\ldots\; P_r \;|\; P_{r+1} \;\ldots\; P_n \;|\; P_{n+1}$$

$$A_1 \quad A_2 \qquad\qquad A_r \qquad\qquad A_n$$

The ratios $\frac{A_n}{A_{n-1}}, \frac{A_{n-1}}{A_{n-2}}, \ldots, \frac{A_{r+1}}{A_r}, \ldots, \frac{A_2}{A_1}$ are therefore established; then, with the use of an airflow function, the pressure ratio at each of the restriction points is obtained. Multiplying all the pressure ratio terms together gives the relation between supply pressure and external atmospheric pressure, or

$$\text{Pressure Ratio} = \frac{P_1}{P_{n+1}} \tag{60}$$

This equation gives the overall pressure ratio required for the system. Table XXX summarizes the restrictions of the various subassemblies within the main gearbox. Thus, for the CH-54A main gearbox, the pressure ratio P_1/P_{n+1} is calculated to be 1.72, and the corresponding pressure drop is

$$\Delta P = \left[\left[P_{n+1} \times \frac{P_1}{P_{n+1}} \right] - P_{n+1} \right] \tag{61}$$

$$= \left[[14.7 \times 1.72] - 14.7 \right]$$

$$= 10.7 \text{ psi}$$

It is further assumed that the air pressure rise through the blower is approximately equal to the pressure drop through the main gearbox.

The relation between temperature, pressure ratio, and power absorption of the cooling air blower required for an oil-fog lubrication system is presented graphically in Figures 30 and 31. In constructing these curves, the following assumptions and basic thermodynamic relations were used:

- Cooling air blower is a high-speed centrifugal unit; therefore, compression is largely adiabatic.

- Compression efficiency η is taken as 70%.

TABLE XXX. AIRFLOW RESTRICTIONS - CH-54A MAIN GEARBOX

Subassembly	Location of Restriction	Area of Restriction (in.^{2})
Rear Casing Cover	A_1 and A_2	20
2nd Stage Planetary	A_3	25
Planetary Interspace	A_4	25
1st Stage Planetary	A_5	50
Interspace Between 1st Stage Planetary and Bearing Support Panel	A_6	30
Bearing Support Panel	A_7	35
Interspace Between Bearing Support Panel and Input Bevel	A_8	50
Outlet Vents in Upper Main Casing	A_9	20
Input Drive Assembly	A_1'	1.0
Free Wheel Unit	A_2'	1.0
Input Drive Vent	A_3'	1.0

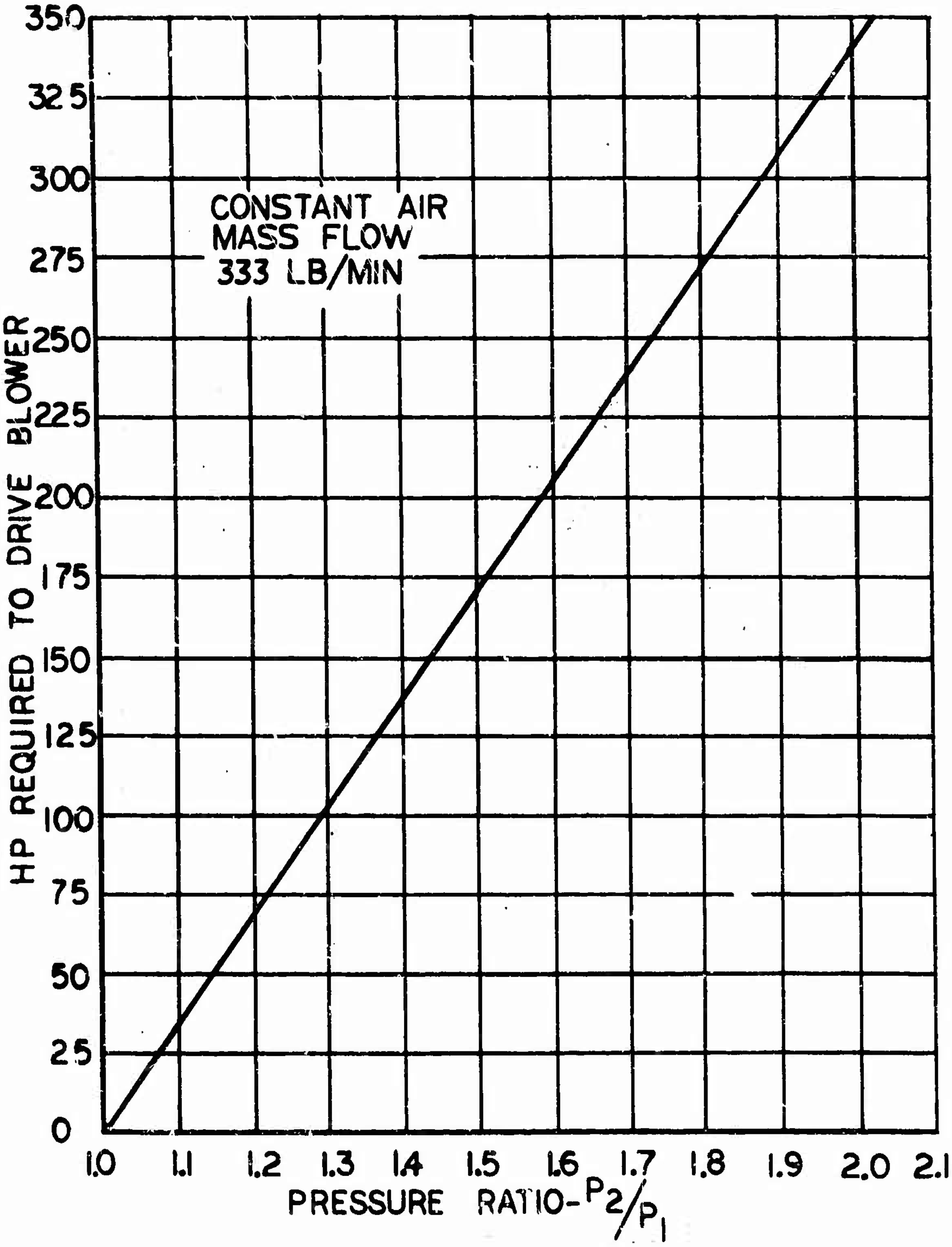


Figure 30. Main Gearbox Cooling Airflow and Blower Power Requirements.

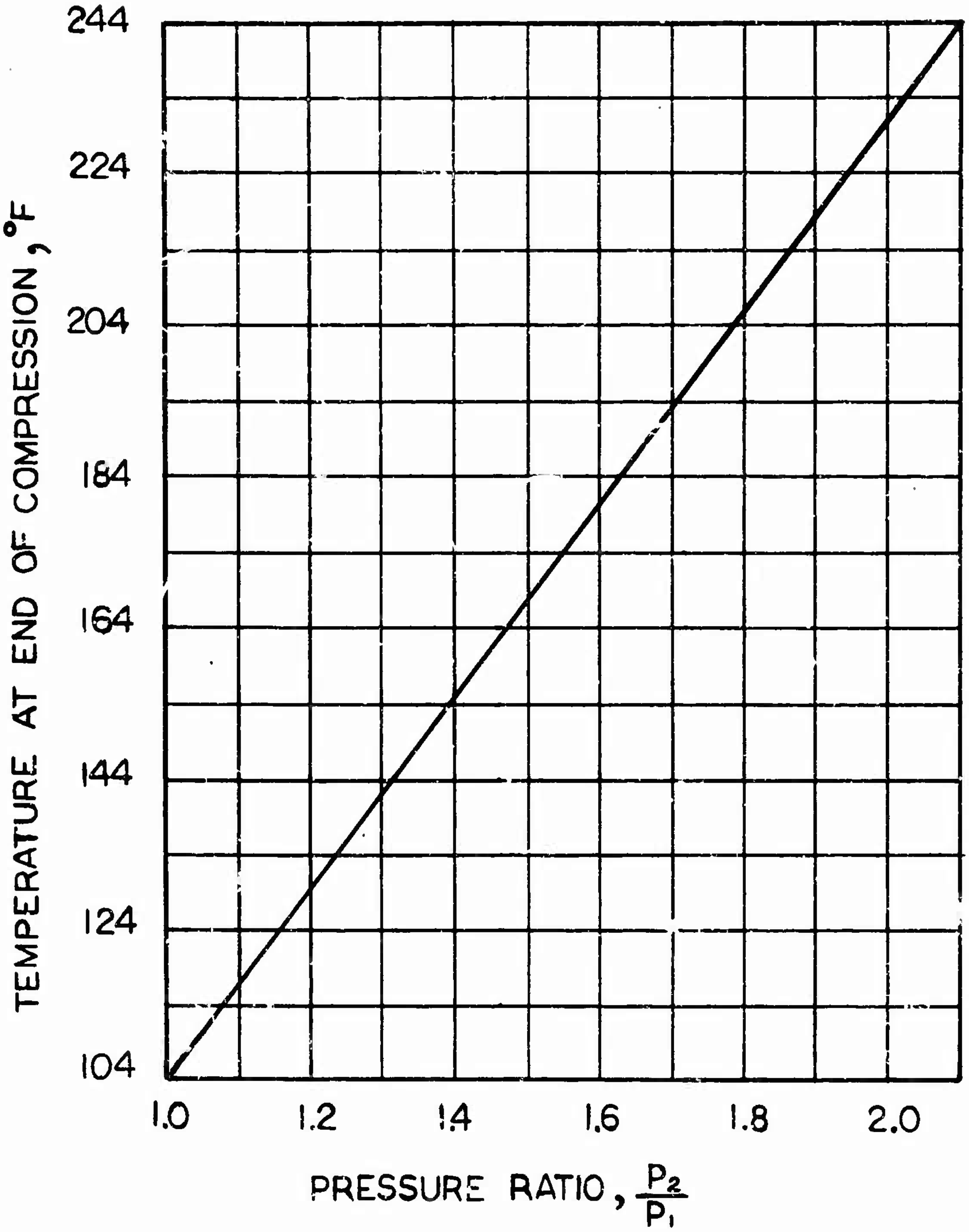
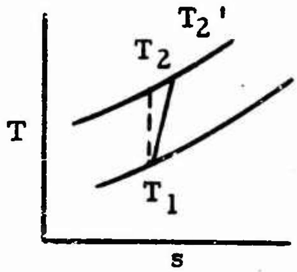


Figure 31. Pressure Ratio Versus Compressor Temperature.

$$\frac{T_2}{T_1} = \left[\frac{P_2}{P_1}\right]^{\frac{\gamma - 1}{\gamma}} \qquad (62)$$

from which T_2, the temperature at the end of an isentropic compression, is found. The power required to drive the cooling air blower is then calculated as follows:



for 100% compression efficiency,

$$\text{work done} = \dot{m}_a C_p \Delta T$$

$$= \dot{m}_a C_p \, (T_2 - T_1)$$

with compression efficiency $<$ 100%,

$$\text{work done} = \dot{m}_a C_p \, (T_2' - T_1) \qquad (63)$$

$$\text{where } T_2' = \frac{T_2 - T_1}{\eta} + T_1 \qquad (64)$$

Substituting equation (64) in equation (63) gives

$$\dot{Q} = \frac{\dot{m}_a C_p \, (T_2 - T_1)}{\eta} \qquad (65)$$

from which the blower HP is

$$HP = \frac{\dot{m}_a C_p \, (T_2 - T_1)}{\eta} \times \text{constant}$$

where the constant is $\frac{778}{33,000}$

For the CH-54A main gearbox, the magnitude of power loss due to oil churning is almost eliminated by incorporation of an oil-fog lubrication system. Referring to Figure 28, at the conditions of maximum transmitted

power, the losses are reduced from 5100 Btu/min to 3100 Btu/min, which is a "saving" of 1900 Btu/min or 44 HP. This power plus the power required to drive the present oil cooler blower (12 HP) and oil supply pump (2.0 HP) is equal to 58 HP. If this power absorption is exceeded by the blower to be incorporated in the oil-fog system, a net loss of power to the helicopter main transmission will occur. Referring back to the estimated pressure ratio of 1.72, which is considered necessary to overcome internal gearbox losses, the power absorption of the blower (as shown in Figure 30) will be 238 HP. This obviously means a loss of power to the aircraft of 180 HP (238 - 58). To obtain a "break-even" in power absorption, the blower would be able to operate with a pressure ratio of only 1.17; i.e., the equivalent power would be 58 HP, as shown in Figure 30. It is doubtful if this pressure ratio will provide a sufficient mass flow of cooling air through the main gearbox.

Cooling Air Blower - Temperature Limitations

The increase in pressure of the cooling air will be accompanied by an increase in air temperature. The following example serves to illustrate that no excessive temperature is encountered at the maximum anticipated pressure ratio.

Considering an aircraft operating at sea level on a 104°F (564°R) day and limiting the outlet temperature of the oil mist system to that proposed for conventional rotor shafts and oil sump cooler systems (i.e., 280°F or 740°R), the allowable temperature rise across the air-oil blower is as follows, where 40° is the mean cooling air rise through the gearbox:

$$\Delta T_{blower} = (280 - 104) - 40$$

$$= 136^\circ F$$

Adding this temperature to the ambient day temperature gives

$$104 + 136$$

$$= 240^\circ F$$

This is the air temperature at the exit from the blower. From Figure 31, it can be seen that the blower could be operating at a pressure ratio greater than (2) without exceeding any temperature limits. Thus, it is demonstrated that power absorption of the cooling air blower is the limiting consideration.

Oil-Fog Supply

A further important aspect with an oil-fog system is to ensure a continuous supply of lubricant under all conditions of operation. The present type of commercially available micro-fog unit is designed for operation in a vertical attitude. There are obviously some limits to which deviation from vertical can be made, without any malfunction. This feature is necessary for successful application in a helicopter, as the pitch and roll motion of the aircraft must not adversely affect the micro-fog unit.

The plumbing of feed lines and "reclassifiers" should not present any major difficulties, but ensuring distribution in the correct quantities to the various components may complicate the installation. A more detailed design study is required to define the installation.

FURTHER ASPECTS OF LUBRICATION AND COOLING

The following discussion highlights some of the major considerations to be made in the use of an oil-fog lubrication/cooling system as a possible replacement for the more conventional recirculating liquid-oil type lubrication system.

Installation Features

The volumetric flow of the blower will dictate the weight of the oil-fog system [illegible]mponents for any given mass flow. If a constant mass flow is assu[illegible] a range of altitude, then the maximum altitude case will de-m[illegible] [illegible]gest volume flow and hence the heaviest unit.

Figure 32 shows this trend; it can be observed that a twofold increase in specific volume can occur between sea level and an altitude of 23,750 feet. Figure 32 indicates that the physical size of the blower should be matched to the aircraft service ceiling, which for the CH-54A is approximately 25,000 feet. It may be possible to incorporate such refinements as a variable-speed drive between the blower and power source (either main gearbox or APU) to vary blower speed and output to compensate for altitude changes. The use of such relatively complicated features not only tends to offset the primary advantage of the oil-fog system (simplicity and light weight) but also to increase the vulnerability to small-arms fire during combat.

Air Filtration

To provide some level of protection against ingestion of dust and sand particles, some form of filtration is necessary with a gearbox cooled internally by air. From the results of work carried out to establish design

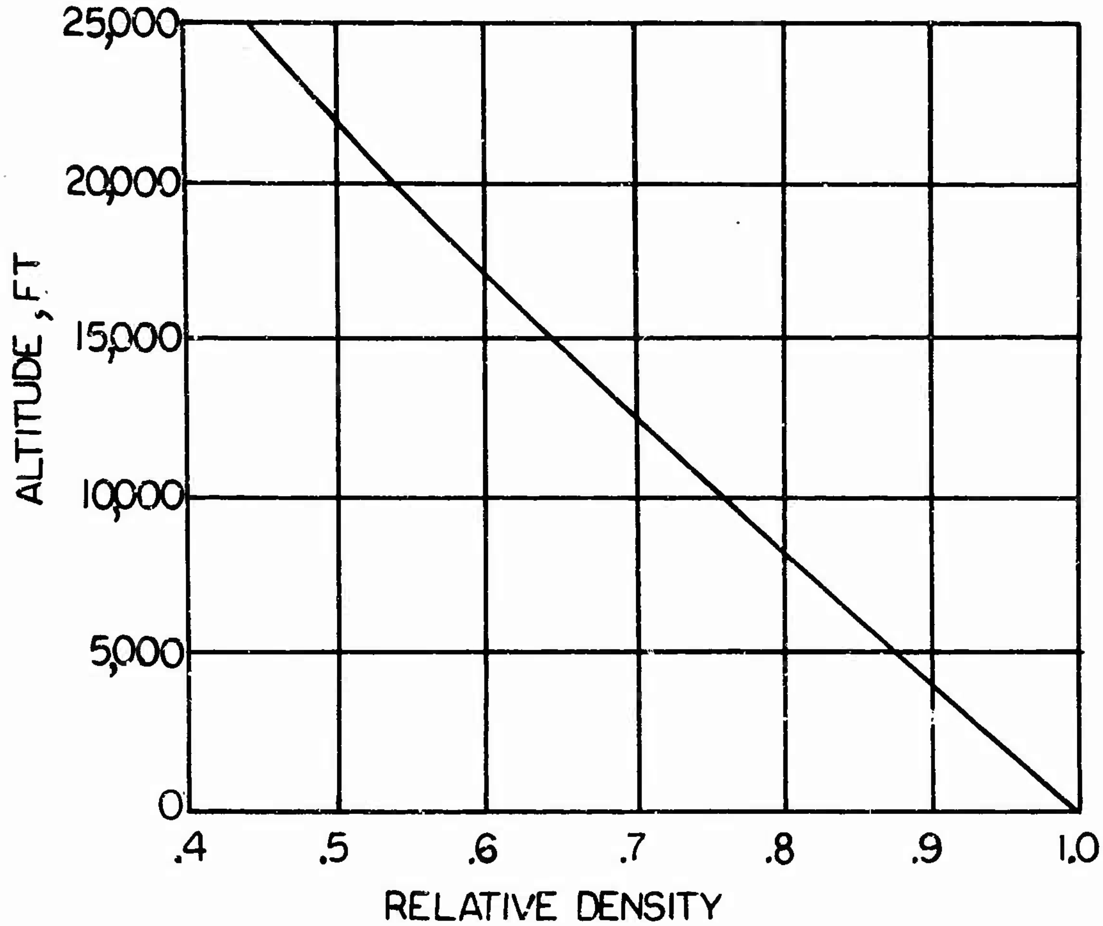


Figure 32. Relative Density Versus Altitude.

criteria for gas turbine air filters, the size of filter for the main gearbox application will be approximately 1 cubic foot.

Air Drying and Water Separation

In a system that uses air for the cooling of precision mechanical components in combination with small quantities of oil, it will be necessary to ensure freedom from moisture entrainment; otherwise, corrosion will occur within the main gearbox. It is felt that the ppm values of water carried in the cooling air during operation in a tropical rainstorm (as may be experienced in Southeast Asia) would be excessive and the water will require separation from the cooling air.

Warning of Mechanical Malfunction(s)

Some equivalent instrumentation is required to give at least the same degree of pilot warning as the present type of recirculating lubrication

systems, which utilize low oil pressure warning lights and chip detectors.

Fire Hazards

The dispersion of oil droplets in a continuum of air represents a possible combustible mixture. Although the anticipated oil-to-air ratios (by weight) would be far from stoichiometric, the possibility of combustion occurring due to contact with a hot surface (i.e., a bearing which has reached a point of incipient failure) cannot be completely dismissed. It is felt that some additional research is necessary in this area.

Lubrication With Minimal Oil Flows

The two major regimes of lubrication that are encountered in helicopter transmissions are:

1. Hydrodynamic "thick" film lubrication

2. Elastohydrodynamic "thin" film (EHD) lubrication

Typical components classified in the first category are sleeve bearings and high-speed thrust washers. To operate successfully, they require an oil flow far in excess of that which can be provided by any micro-fog system; the system therefore cannot be recommended for any application incorporating such components. Components which operate with EHD lubrication are essentially gears and rolling contact bearings. Gears operate with heavily loaded rolling plus sliding motion and from preliminary investigation would appear to have a reasonable chance of successful operation with low oil flow, if adequate cooling is provided. With rolling contact bearings, however, the possibility of utilizing micro-fog with any confidence appears marginal, since there are areas within the cage (retainer) pockets and location lands that operate with almost pure sliding motion. As an example, a ball bearing with an inner centered cage may have a radial clearance of only a few thousandths of an inch with a relatively large L/D ratio. Under the adverse conditions of high speed and combined thrust and radial load, large radial forces can exist between the cage bore and its locating land. To function effectively, the bearing must operate similar to a plain bearing and will require the larger oil flows.

Referring now to the data obtained from testing with the Ryder gear tester, Figure 33 shows a plot of the percentage of an assumed value of friction horsepower absorbed into the cooling air against the transmitted horsepower. It is difficult to draw any firm conclusions, as more than one variable has been altered at different conditions. Obviously, much more testing will be required with both gears, and more so with rolling contact

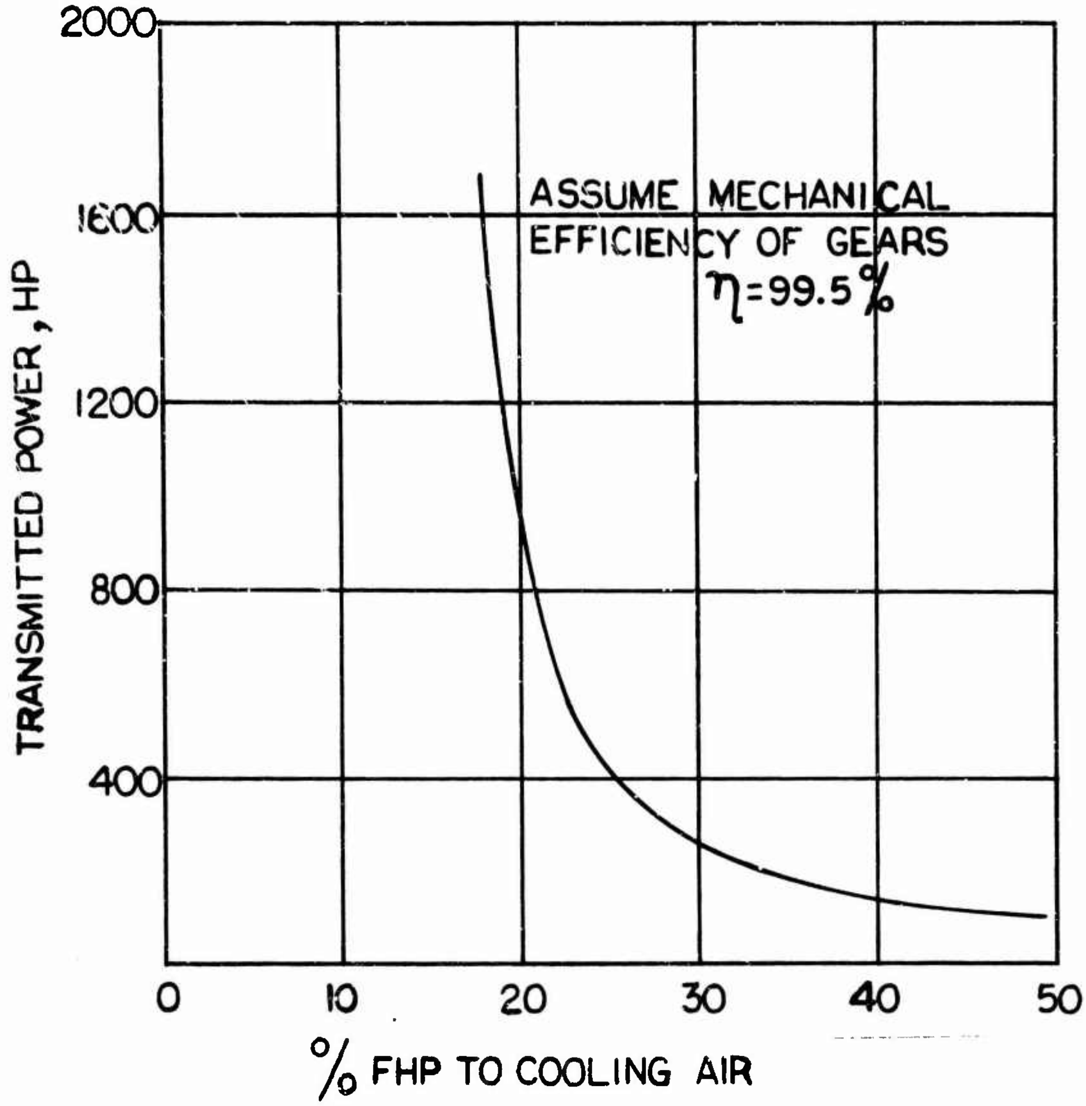


Figure 33. Percentage of FHP to Cooling Air Versus Power Transmitted.

bearings, whereby test programs do not idealize running conditions but are arranged to simulate those experienced in service operation.

CONCLUSIONS

The findings of this study suggest that an oil-fog (micro-fog) lubrication/cooling concept should not be recommended for immediate use in a high-speed reduction gearbox, as used in a helicopter transmission system. The changes that become necessary in the use of any such new and untried system will dictate further research and testing to establish new design criteria.

Oil-fog lubrication is not recommended for use in components such as highly loaded sleeve bearings which need to operate with hydrodynamic "thick" films (hence, high oil flows) to ensure efficient performance.

The lubrication of spur gears by oil-fog has provided some interesting results. These tests were conducted with gears of relatively fine pitch as compared with the coarse pitch gears used in helicopter gearboxes. Further testing with a coarse pitch (3 or 4 DP) gear set would yield valuable data as more demands would be placed on the lubricant, since the gross loaded sliding within the tooth contact is more severe.

No published information has been located which describes any test data on large (1×10^6 DN) high-speed bearings lubricated by oil-fog. Much test data concerning the uses of new materials and low friction coatings for bearing cages (retainers) is required if any future application is to be made of oil-fog lubrication techniques.

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13. ABSTRACT

This report presents the results of a feasibility and preliminary design study of integral cooling systems for helicopter main transmissions. The purpose of the study is to provide a method or methods whereby the requirements for remote external cooling systems no longer exist. The goal of the program is to define long-range solutions and/or expedient methods permitting simple retrofit to existing aircraft. While the primary effort centers around the heat rejection requirements of the main gearbox of the U.S. Army CH-54A crane helicopter, the adaptability of the most promising systems to other U.S. Army helicopters is also investigated. Several extremely advanced methods of gearbox cooling were investigated. Although the thermodynamic principles of these concepts are sound, current state-of-the-art hardware makes the systems impractical because of weight and effective power requirements, which result in significant degradations in aircraft performance. Two of the systems investigated, the rotor shaft and oil sump coolers, present relatively efficient and invulnerable solutions to the problem. These systems are substantiated by detailed analysis, empirical data, and consultation with many heat exchanger manufacturers. A vapor cycle refrigeration system is also evaluated. Detailed reliability, maintainability, and vulnerability analyses of the three systems are presented in comparison to the present CH-54A external cooling system. In addition to a failure mode and effect analysis, estimated maintenance man-hours and MTBFs are presented for the integral cooling and current external cooling systems.

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